Eglin 1
Housing and Deployment Design for an Acoustic Eye Sensor

Sponsor: Eglin AFRL/MNGN
Henry Pfister PhD.

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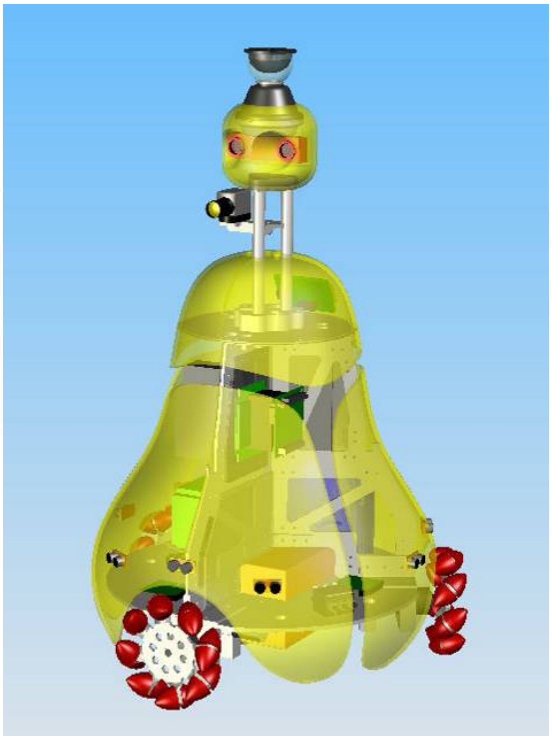

EML 4551 Senior Design Project
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April 4, 2006

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Introduction

The Eglin 1 senior design team, design for a housing of a tetrahedral acoustic array sensor, consists of Erik Fernandez, Kevin Garvey, William Heffner, and Brian McMinn at Florida State University. This project is in coordination with Dr. Henry Pfister and the Air Force Research Lab (AFRL/MN) located on Eglin Air Force base. The scope of this project is to develop a foldable housing for a tetrahedral microphone sensor array that was designed and developed by Dr. Pfister. The housing itself is to be attached to a Robotic Demonstration System (RDS) that is being built by NASA Langley. The conjoined efforts of the Eglin 1 senior design team and Dr. Pfister will ultimately produce an acoustic navigation system that NASA Langley can use for their prototype robot. A secondary objective was also added in the later phases of the project. This addition was to design and build a half-size, non-collapsible, T-base array that would be implemented on a VEX™ robot. VEX™ robots are simple robot kits distributed by any Radio Shack® store and will be utilized by Eglin AFRL/MN to test the acoustic eye utilizing a smaller array size.

Background

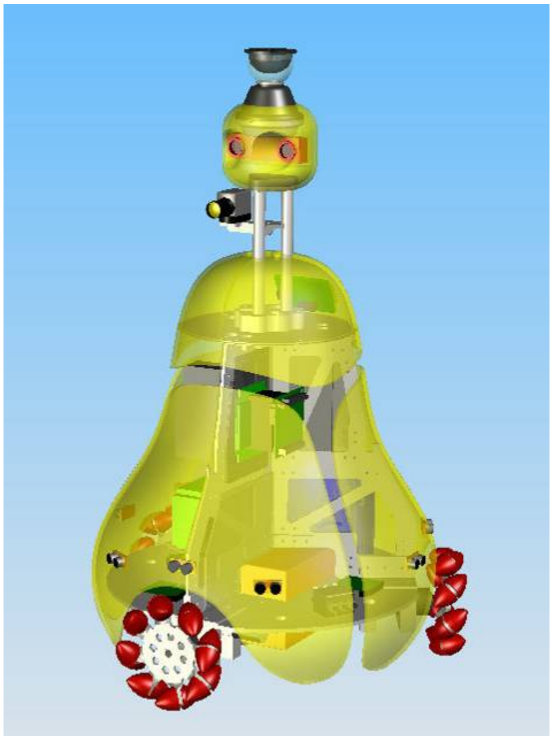
AFRL/MN is working on an innovative Robotic Demonstration System (RDS) designed by NASA. The RDS will be equipped with a variety of sensors, one of which will be an Acoustic Eye. The majority of the Senior Design Project will center on designing a collapsible, vibration damped housing for the Acoustic Eye sensor. Secondarily, a T-base array will also be designed to accommodate the testing that AFRL/MN will do on the VEX™ robot. The Acoustic eye sensor will aid the RDS in navigating in cluttered environments by utilizing a tetrahedral four microphone array that will use sound signals to determine position and elevation from a sound source. The primary project goal is to design and build the actual tetrahedral frame for the sensor. The secondary goal is to make this housing vibration resistance and damp out mechanical noise caused by the RDS.

The purpose of this tetrahedral array frame is to contain the acoustic sensors which will be used to navigate the robot. Each sensor will contain a microphone, placed 20 inches apart in order to determine where the robot is located with respect to its surroundings. The microphone sensors send their respective signals to the RDS computer where it is processed into an algorithm where the calculations are made for the robot to determine its orientation to the source.

The purpose of the T-base array frame is to accommodate an identical acoustic eye sensor to the tetrahedral array frame, only on a smaller scale. The scale of the T-base array is exactly on a ½ scale to the tetrahedral array frame. The microphones will be placed at 10 inches to one another and utilize a separate processor and algorithm than that of the RDS robot. This smaller array will serve to provide a test bed for the acoustic eye until the RDS robot from NASA comes online.

What is the RDS (Robotic Demonstration System)?

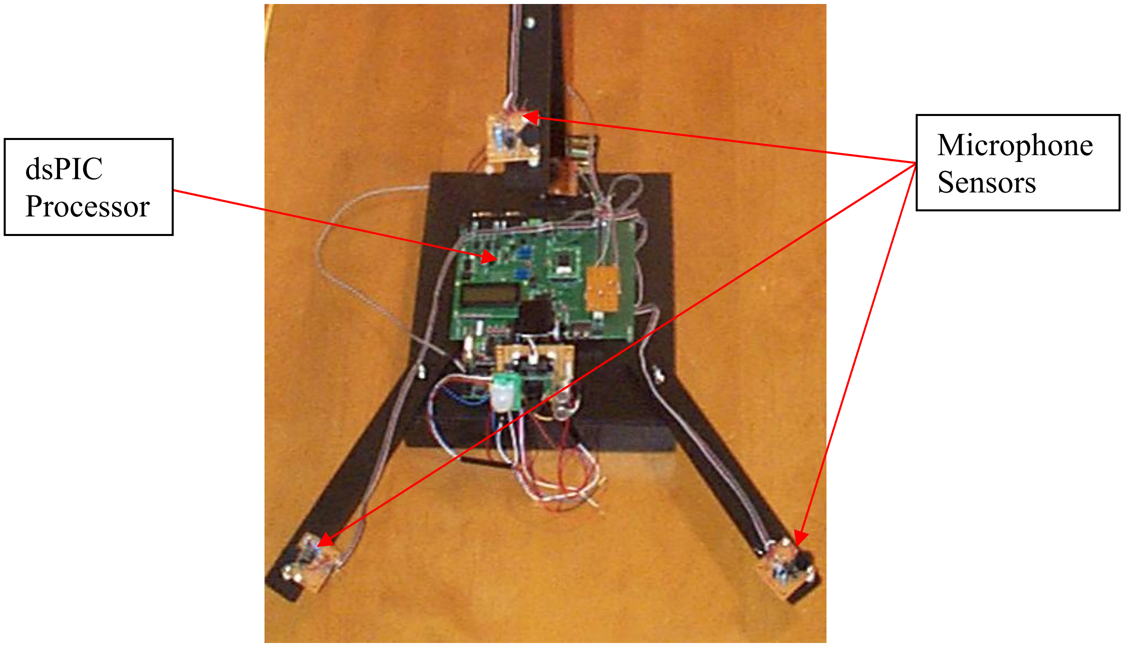
The RDS or (robotic demonstration system) is a robotic platform used for demonstrating integration of multiple computers, different types of sensors, and other device types. The purpose is to educate the public in different types of upcoming technologies. An example of the RDS is depicted below in (figure 1-RDS) and includes a dsPIC processor, omni-directional camera, pan and tilt sensors, IR sensors, and an acoustic eye sensor. The main scope of the senior design team's task centers on integrating a vibration damped housing for the acoustic eye sensor into the RDS platform.



(Figure 1-RDS)
(NASA RDS Platform)

What is an Acoustic Eye Sensor?

The acoustic eye sensor being integrated into the RDS platform is a microphone array setup that utilizes sound signals to "see" and process location and elevation from the sound source. The acoustic eye sensor used in the senior design project can be seen in (figure 2-Acoustic Eye) and uses four of the microphone sensors setup into a tetrahedral array. The sensor uses a dsPIC processor and algorithm to calculate the position and elevation of the sound signal from the entire array of microphone sensors. The sensor works by calculating the time difference that a sound signal takes to reach the four individual microphones sensors within the tetrahedral platform. The senior design team was tasked with designing a vibration mitigation housing to integrate the acoustic eye sensor into the RDS platform.



(Figure 2-Acoustic Eye)
(Photo Courtesy Dr. Pfister, Eglin AFRL/MN)

Constraints

Several constraints have been set forth for the overall design process and had to be considered and ranked before any design work began. The overall constraints were ordered such that the most important problems were addressed first. Constraints that were left open-ended by AFRL/MN were discussed further to achieve an agreement as to how the senior design team would proceed. The constraints are listed out below from the most important to the least important.

Tetrahedral Array Constraints

1) The microphones connected to the array must be oriented in a tetrahedral shape.
2) The center-to-center distance of the microphones must be located at a distance of 20 inches from one another on the apices of the tetrahedron frame.
3) The tetrahedral frame cannot interfere with any existing sensors already equipped on the robot except for the omni-directional camera.
4) All microphones must be facing upward, and lie in the horizontal plane.
5) The design must be collapsible (manually or autonomously) and adaptable to the robot.
6) The tetrahedral frame should damp out as much mechanical vibration as possible.
7) When collapsed, the tetrahedral frame should not protrude outside the overall footprint of the RDS robot. (This is approximately 13 inches, as NASA has yet to finalize their design).
8) The overall height of the tetrahedral frame should add no more than 25% of the existing RDS robots height. (This is limited only by the mounting location for the frame to the robot).
9) The design must be cost effective and as cheap as possible.
10) Use as many off-the-shelf components as possible.

T-Base Array Constraints

11) The microphones connected to the array must be oriented in a tetrahedral shape.
12) The center-to-center distance of the microphones must be located at a distance of 10 inches from one another on the apices of the tetrahedron frame.
13) The tetrahedral frame cannot interfere with any existing sensors already equipped on the VEX™ robot.
14) All microphones must be facing upward, and lie in the horizontal plane.
15) The design must be adaptable to a standard nail plate supplied by ACE Hardware®.
16) The tetrahedral frame should damp out as much mechanical vibration as possible.
17) The array must be as lightweight as possible so as to not impose large loads on the robot frame.
18) The design must be cost effective and as cheap as possible.
19) Use as many off-the-shelf components as possible.

Background Research

The constraints listed above had to be analyzed to understand where the project would be centered. From the constraints presented by AFRL/MN and Dr. Pfister, the most important aspects of the project were outlined and researched. The main scope of the design process was determined to be a materials analysis to combat the effects of possible mechanical vibrations reaching the microphone sensor.

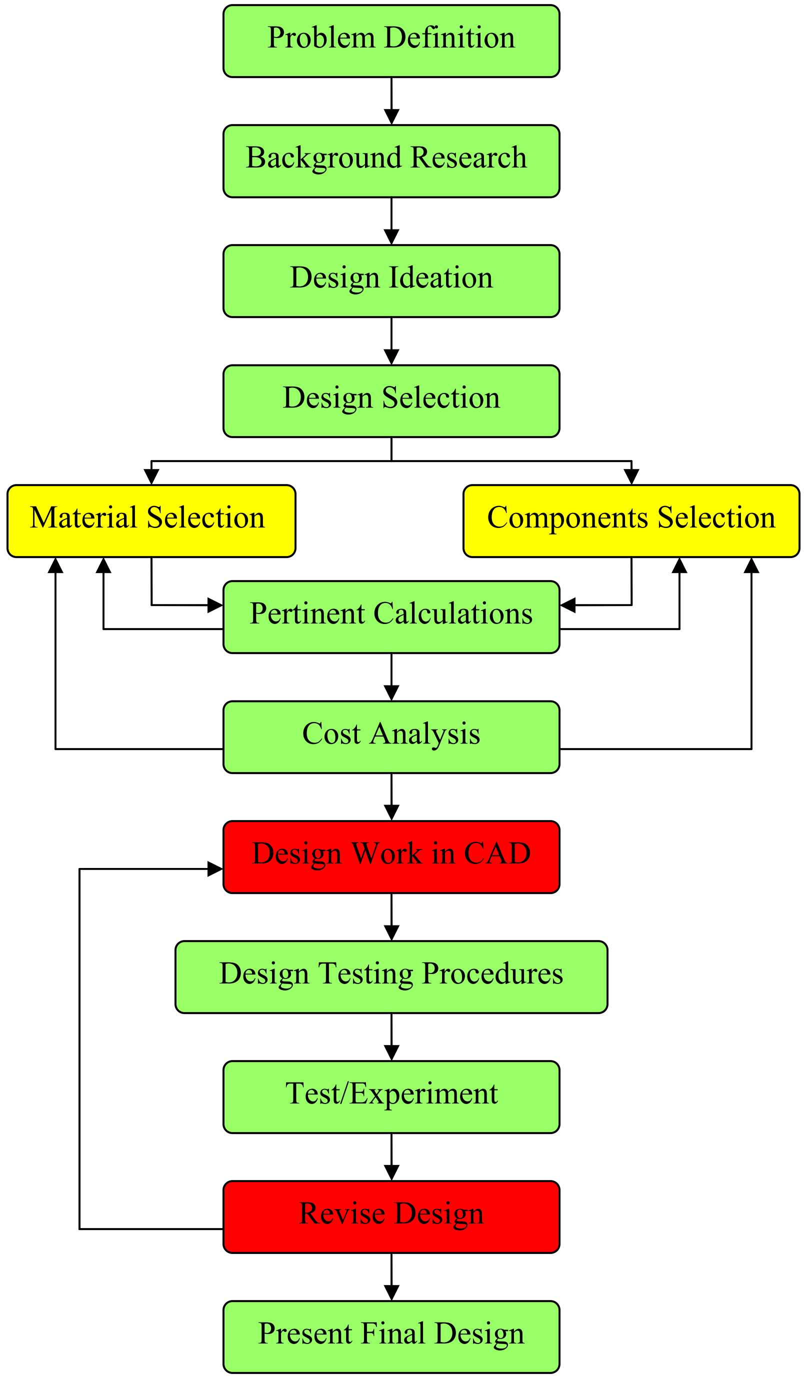
The microphone sensor has a signal range of up to 20 kHz which is sent to a dsPIC controller on the RDS robot. The dsPIC digitizes the analog signals from the microphone sensors at 20 kHz per channel. The tetrahedral microphone array will include four such microphone sensors which will require processing by the dsPIC controller. The main issue with the current microphone sensor array is that it is receiving mechanical vibrations from the structure to which it is mounted. This is where the scope of the project for the senior design team is centered.

Materials required to damp out the mechanical vibrations throughout the structure were researched from the standpoint that the structure must be cost effective and lightweight. Different families of materials were researched such as foams, polymers, natural materials, metals, and composites. The research showed that metals and composites conducted vibration far too well. Polymers, foams, and natural materials had better characteristics in terms of vibration conduction through the material and further research was decided upon for these materials. The research into these material families was narrowed even further and is outlined in the (**Materials Selection**) section of this document.

The remaining research for this project centered on determining the availability and cost of the other components required to assemble the structures once a material was chosen. Cost effective, off-the-shelf hardware was sourced from different suppliers to determine feasibility of their use for the design phase. Once suppliers were determined, conferencing began as to their specific product line and how their specific product could be implemented in the design phase. The different suppliers chosen were Sorbothane®, Igus®, Lowe's®, and McMaster Carr®. The specific research into their specific components is presented in the (**Components Selection**) section of this document.

Project Plan

The scope of the project presented by Eglin AFRL/MN and Dr. Pfister was to develop a tetrahedral acoustic eye sensor housing that mitigated mechanical vibration while being a foldable platform as well as a smaller secondary non-collapsible array with the same damping characteristics. The constraints laid out above in the (**Constraints**) section were analyzed and research done so that a project plan could be devised. The project plan is depicted on the next page in the flow chart (figure 1-FlowChart).



(Figure 1-FlowChart)

Design Ideation

Background

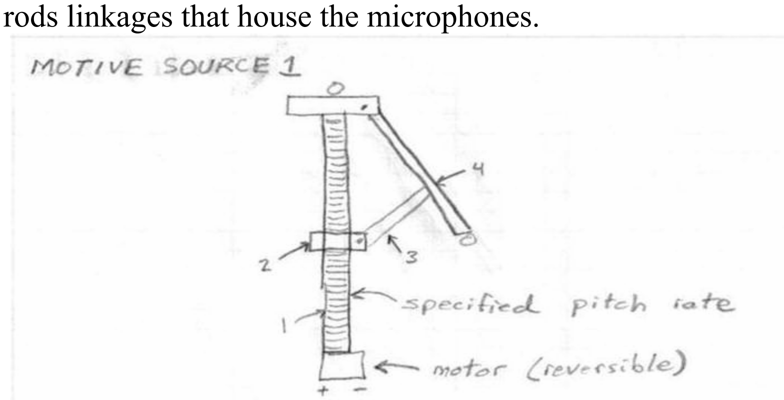
Several ideas had to be generated that could have the potential to solve and adhere to the constraints outlined by the AFRL/MN. The designs generated vary in their deployment method, rigidity, cost, complexity, and weight. These various designs were evaluated with regards to their advantages and disadvantages, and an optimum design selected.

Design Concepts

Throughout the process of considering different designs, many apparent derivatives to existing designs came forward. Narrowing the plausible designs from the list of design ideas generated the four main designs that seemed to adhere to the constraints from AFRL/MN the best for the tetrahedral array frame. As for the T-base array, only one design is outlined because the design itself was provided by AFRL/MN. These designs are outlined below and provide evidence as to how each design may be implemented to complete the project goals.

<u>Screw-Type Tetrahedron over Tetrahedron frame</u>

Implements a design with two tetrahedrons mounted base to base. This design uses a screw powered center shaft driven by an electric motor. As the screw is operated the external links are deployed, moving the microphones into position. Only 4 links are required to accomplish this design as seen in (figure 1-Idea1). The mechanism's speed will be based upon the pitch of the screw and the speed of the motor, which will be determined to accommodate the type of material utilized for the design. The turning screw will have a threaded collar, attached to the center screw shaft, which will operate the extension rods linkages that house the microphones.



(Figure 1-Idea1)

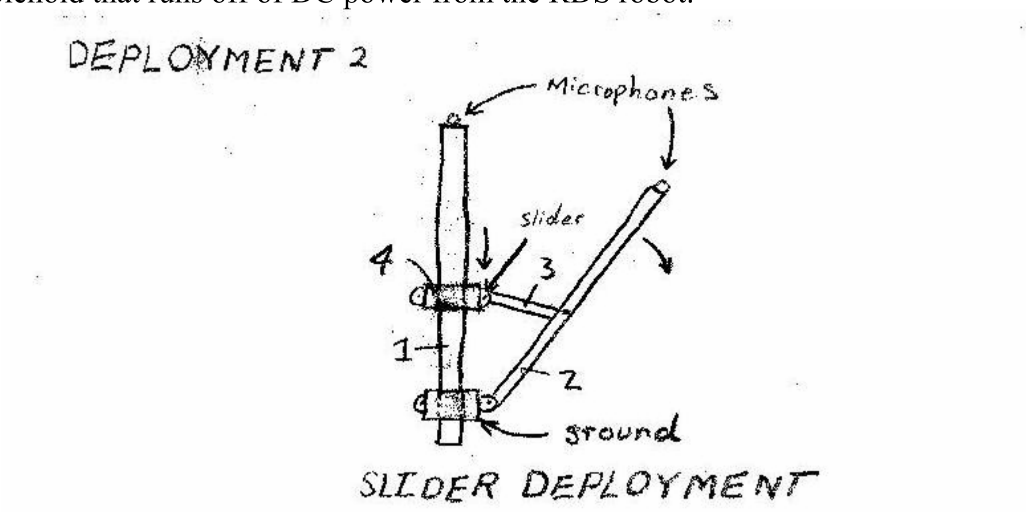
The advantages to this design are as follows:
- Requires very few parts for operation (only 4 links for each reference plane)
- Simple to design and build
- Requires no locking mechanism to hold microphones in place when deployed
- Ranges of motion and deployment speed are adjustable

The disadvantages of this design are as follows:

- More expensive design due to the screw-type mechanism
- A force is imposed on the center shaft teeth that will require some sort of lubrication to warrant off wear and friction

Four-bar slider frame

The four bar slider linkage will be a tetrahedral array that can deploy up or down as in (figure 1-Idea2). The center support shaft will house the slider link. The bottom main base in which the three arms are attached will serve as ground. The arms are connected to a slider link, which will slide up and down the support shaft. When the slider translates to its opposing position the linkage is fully deployed. To fold the array, the slider simply retracts and the arms are fully drawn back into their closed position. For automation purposes, the means for deployment for this frame will need to be an actuator or solenoid that runs off of DC power from the RDS robot.



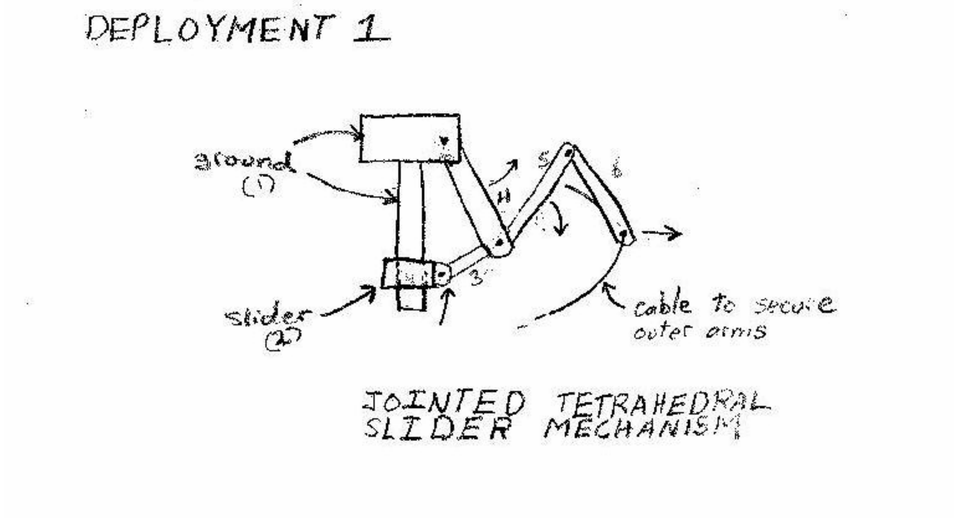
(Figure 1-Idea2)

The advantages to this design are as follows:
-Only three moving parts (Slider Link, Connecting Arms, and Extension Rods)
-Simple to design and build
-Simple to convert to a fully automated deployment system
-Deploys to fully deployed position with the movement of a single link
-Inexpensive
-Can accommodate several mounting positions on RDS Robot

The disadvantages of this design are as follows:
-Must be constructed with tight tolerances to minimize vibration between links
-Moving parts cause friction, which leads to wear
-Material selection is crucial in this design since the arms cannot be allowed to flex (change in microphone to microphone distance is not acceptable)

Jointed Tetrahedral Slider Mechanism frame

The mechanism in (figure 1-Idea3) will consist of a main base ground link and a lower collared slider link. As the slider approaches the cap of link 1 it pushes link 3 into link 4. Link 4 will pivot upward, which extends the outer links away from the center. Using a cable to maintain the proper position of the outer extended position of link 6, the microphones will be deployed into their proper operating position.



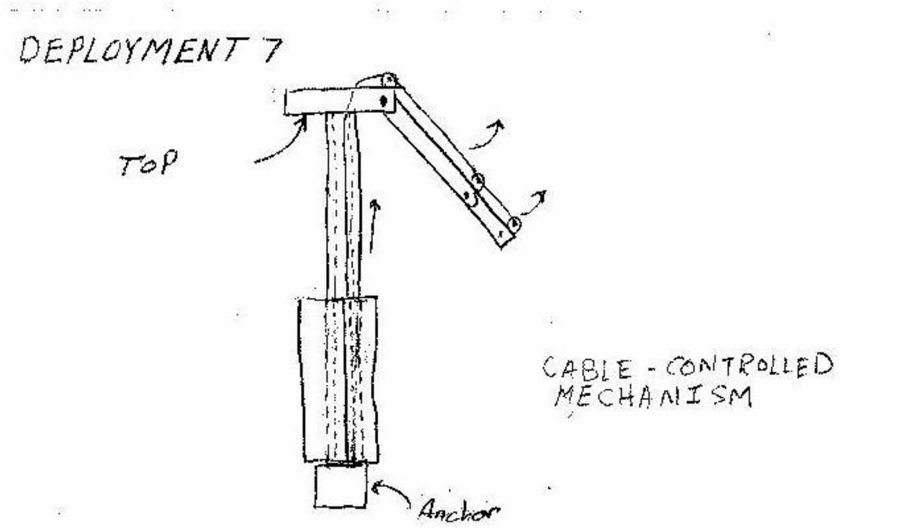
(Figure 1-Idea3)

The advantages to this design are as follows:
-Input of force is isolated to the slider mechanism.
-Extra cable provides extra rigidity to frame when fully deployed.
-Simple to convert to a fully automated deployment system
-Deploys to fully deployed position with the movement of a single link
-Extremely compact size

The disadvantages of this design are as follows:
-Must be constructed with tight tolerances to minimize vibration between links
-Moving parts cause friction, which leads to wear
-Material selection is crucial in this design since the arms cannot be allowed to flex (change in microphone to microphone distance is not acceptable)
-Excess linkages in design can cause more vibrations than other designs.
-More complex structure allows more opportunities for frame to malfunction
-Extra linkages make design development more difficult

Cable Controlled Deployment Mechanism frame

The cable-controlled mechanism is based on a hollow center tube design. A small cable is run from the base of the frame through a hollow ground tube and down each individual arm. As the cable is tightened, the arms extend outward to their deployed position because of the tension produced in the cable. The motion of the arms is limited by having the range of each joint limited in motion to allow full deployment. The cable tightens either by forcing or releasing the outer hollow tube (with cables attached) upwards or downwards over the inside tube in effect pulling the cable. This action will cause the cables to deploy the extending rods to be deployed as in (figure 1-Idea4). To automate this system a motor may be attached to the cables instead of the outer hollow tube and the motor could simply retract the cables into a spool at the base.



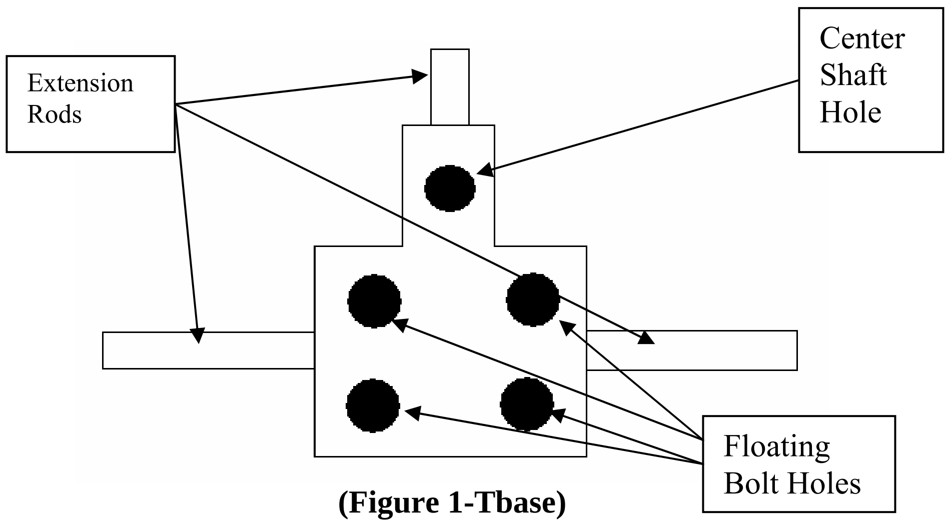
(Figure 1-Idea4)

The advantages to this design are as follows:
-Moderately simple to construct
-Force is applied through one line of motion
-Simple to design and build
-Simple to convert to a fully automated deployment system
-Deploys to fully deployed position with a single driving device
-Inexpensive
-Can accommodate several mounting positions on RDS Robot

The disadvantages of this design are as follows:
-Must be constructed with tight tolerances to minimize vibration between links
-Moving parts cause friction, which leads to wear
-Cable may fatigue over time, leading to malfunctions
-Arms are not held extremely rigid, which may cause excess vibration
-Locking mechanism may be needed for the extension rods once deployed
-Material selection is crucial in this design since the arms cannot be allowed to flex (change in microphone to microphone distance is not acceptable)

T-Base Design

The T-base design consists of a very basic idea as outlined by its name. The base of the array is actually in a T configuration as seen in (figure 1-Tbase). The design calls for a tetrahedral setup with microphones at the apices of the tetrahedron form. This design constitutes three extension rods that form the base of the tetrahedron as well as a center shaft that produces the apex of the tetrahedron. This design was provided by AFRL/MN and minor modifications were the only design work implemented on the final T-base.



(Figure 1-Tbase)

Final Design Selection Process

The final design selection was made after reviewing all the pertinent information and implementing them into a design matrix. The design matrix utilized focuses on four of the designs generated. The four designs that showed the most promise were the tetrahedron over tetrahedron screw-type, the four-bar slider folding down, the jointed tetrahedral slider folding up, and the cable-controlled deployment. The following is the design matrix (table 1-Design1) implemented in the determination of the final design concept selection and following the matrix is the explanation behind how and why the particular methods were utilized.

Mechanism	Price to Produce (0.2)	Weight (0.15)	Rigidity (0.05)	Complexity (# of parts) (0.1)
A	4	4	8	5
B	8	5	6	5
C	7	2	6	5
D	1	4	3	1

Adaptability (0.25)	Machinability(0.05)	Vibration Resistance (0.15)	Overall Height (0.05)	Total
4	4	2	10	0.481
8	7	5	10	0.9
3	5	2	10	0.3875
2	8	3	10	0.406

Key	
A	Tetrahedron over tetrahedron screw-type
B	Four-Bar Slider Linkage folding down
C	Jointed Tetrahedral Slider
D	Cable Controlled Deployment

Ratings	**1**	**5**	**10**
Price	Expensive ($175+)	Mid-Range ($140+)	Cheap (>$100)
Weight	Heavy (+10 lbs)	Moderate (+5 lbs)	Light (>5 lbs)
Rigidity	Soft (Very Flexible)	Firm (Moderate Deflection)	Rigid (No Deflection)
Complexity	17+ Parts	15-17 Parts	>15 Parts
Adaptability	Complex	Moderate	Simple Mounting
Machinability	Complex	Mid-Range	Simple
Vibration Resistance	Poor	Moderate	Outstanding
Overall Height	<25% Taller than Robot	Up to 25% Taller	Up to 10% Taller

(Table 1-Design1)

Findings

The design matrix above shows that the best design is the four-bar slider linkage folding down, with a weighted value of 0.9 out of 1. This design generates the best performance in terms of accomplishing the goal considering the constraints of the project. Through group meetings and conversation with Dr. Pfister about the requirements for the project; it was determined that the most important constraints are as seen in (Table 1-Design1) above. The design matrix is on a scale from 1 to 10, with 1 being the worst and 10 being the best. A weighting process was then implemented to normalize all the constraints based upon there importance in the design. The aspects behind each rating in the matrix will be discussed in full by each category in the following paragraphs.

Price (0.2)

The price rating was determined by researching different components and materials that fit the design criteria and determining how many parts each design required. The weighting factor for the price was determined to be 20% after much consideration over the other constraints. The calculations for the approximate price for each design can be found in Appendix A. From these calculations a price range was determined and implemented into the format seen in **(table 1-design1)**.

The prices were calculated utilizing components researched from McMaster-Carr®. The component part numbers documented reflect the preliminary choices that have desirable traits for the designs generated.

Weight (0.1)

The constraint for the housing's weight was determined to be no more than 10 pounds not including electric motors or solenoids. The weighting value for the weight of the housing was determined to be 10% as it is not as important as price and adaptability. From this preliminary value, the group decided that the housing should weigh no more than 5 pounds. By looking at the different components that make up each design and the amount of each component used, an estimated weight was determined for each design. The rating for each of these weights was then placed against the range in the design matrix and ranked accordingly.

Rigidity (0.1)

The rigidity constraint was determined based upon the fact that the microphone spacing is a very delicate constraint. The weighting factor for the rigidity was determined to be 10% of the overall constraints due to the fact that some flexing is needed to aid in the vibration damping characteristics of the structure. The microphones are required to be 50 cm apart with a tolerance of 2 cm from the centerlines of each microphone. The rigidity of the housing determines how well this tolerance can be met. The ranking system was determined by looking at each design and how it would be implemented and built and determining which would be more or less rigid based against the other design options. The material choice used in each design is the same and therefore the rigidity now only becomes a factor of how each design is assembled.

Complexity (0.1)

The complexity of each housing design is based upon the number of parts each design utilizes. The weighting factor for the complexity of the housing was set at 10%

because as the design progresses and evolves it may become simpler or it may become more complex; the latter is not a concern for the overall project goal. This simple calculation just involves adding up the amount of each component in each design as seen above in figure 2.

Adaptability (0.25)

Adaptability is based upon how well the housing design can mount to the robot and whether or not there is interference to existing sensors on the robot. The weighting factor for the adaptability was determined to be 25% because it is the most important design parameter. The scope of the project is to develop a housing that is adaptable to the already existing robot design making adaptability the most vital of all the constraints. The ranking was determined by the actual geometry of each housing design and the design matrix exhibits how each design stacked up in this respect.

Machinability (0.05)

The machinability is based upon the simplicity and number of parts that will require machining for each design. The weighting factor for the machinability of the housing components is not of major concern so it only gets a weight of 5% compared with the overall constraints. The ranking system reflects the magnitude by which each design fares against approximate machining and machine shop time required.

Vibration Resistance (0.15)

The vibration resistance of each device was ranked according to the number of moving parts as well as number of joints. The weighting factor for the vibration resistance of the housing was determined to be 15% due to the fact that one of the design constraints is to vibration damp out mechanical noise and this constraint is considered a secondary directive. The more moving parts and joints required, the less the vibration resistance of the device. The simpler the design, the more readily available the vibration solutions are possible. This in turn determined the ranking system seen in the design matrix.

Overall Height (0.05)

The overall height of the design housing had specific constraints on the project. The overall weighting factor for the housing height was determined based upon the idea that the height of the housing was to be an estimated constraint. The weighting is 5% for the overall height because this constraint can be evolved over the time of the project as the robot geometry evolves. The design geometry is such that the housing cannot be any more than 25% of the existing robot's height. The actual geometry required to implement each design determined the ranking system and where each design fell within the parameters set forth in the design matrix.

Final Concept Selected: Four-Bar Slider Frame

The design matrix yielded the four-bar slider frame that folded down as the best design choice for implementation. The design of the tetrahedral four-bar slider frame went through an evolutionary process throughout the design phase such that the best design and materials possible are implemented for the final proposal to AFRL/MN and NASA. The current design is shown in (figure 1-Tetrahedral).

(Figure 1-Tetrahedral)

Four-Bar Tetrahedral Frame Material selection

Purpose

In selecting the material for the four-bar slider housing there were four major considerations for the design. These were weight, availability, price and the speed at which sound and vibration traveled through the material. Since the design called for low weight and poor vibration conduction, the first group considered were metals. These were immediately eliminated since they had high densities and conducted vibration very well. The next group of materials that could possibly meet the constraints was polymers, rigid foams, and composites. Rigid foams were eliminated since they are not readily available, but more importantly their very low densities and higher strengths make them good vibration conductors which is the opposite of what the design requires. Composites were eliminated due to their excellent vibration conduction, which is comparable with metals.

Process

A list was then compiled using primarily polymers (carbon and graphite are also in the final list due to their availability) from the Material Selection in Mechanical Design book by Michael F. Ashby. Polymers have all the properties that are crucial for the tetrahedral four-bar slider frame design chosen. They are easily available in tube, rod, and disc form; are very inexpensive, lightweight, and are poor vibration conductors. Since most of the polymers met the first three criteria the deciding factor would be the speed at which sound (vibration) actually traveled through each material.

Requirements

The primary factor determining which materials will be used for the tetrahedral four-bar slider frame is the speed of sound (vibration) through the material. The lower the speed at which the longitudinal wave travels through the material, the better suited it is for vibration damping. More energy must be exerted by the wave to travel through the material. By using a material with low sound conduction the vibration is damped further by the material itself, therefore vibration damping devices and substrates are not the only forms of vibration damping in the design. By utilizing a material with poor sound conduction, the structure itself will aid in the vibration damping by causing the vibration sound source to lose energy through the frame as the material oscillates minutely. All the materials up to this point have been selected on the basis of price, availability, and low density to minimize the weight of the structure. The final material selected for the frame will posses all of the above properties but will also posses superior vibration damping capabilities. The sound conduction resistance was calculated utilizing the equation in (figure 1-Material1).

The speed of sound in a solid

$$\nu = \left(\frac{E}{\rho}\right)^{\frac{1}{2}}$$

Where E is Young's modulus and rho is the density of the matrial.

(Figure 1-Material1)

Selection Method

The selection method used to choose the materials in the frame design was determined by the sound velocities. It is clear that UHMW-PE (Ultra High Molecular Weight-Polyethylene) is the worst sound (vibration) conductor of all of the final materials considered in (Table 1-Material2). In this case, worse sound conduction is better for this application because we need to damp out mechanical vibration and not conduct it. **From these numbers, the final decision can be made that UHMW-PE is the best material for this deployment mechanism.** ABS (Acrylanitrile butadiene Styrene) plastic is the second choice and was chosen for certain components in the design for availability reasons. Not only does UHMW-PE fulfill the strength and weight requirements, but it also conducts sound (vibration) poorly. This material property will greatly aid the vibration damping of the tetrahedral four-bar slider frame.

The following table (Table 1-Material2) illustrates how all of the final materials compared with each other and which ones had lowest wave speeds.

Materials Considered	Price of Material ($/ft) for 1 inch diameter	Density of Material (lb/in^3)	Speed of Sound Through Material (ft/s)
Polycarbonate	4.32	0.044	4415.82
ABS	5.2	0.04	4254.163
Carbon	13.8	0.0813	4791.61
Graphite	37.83	0.0643	9364.501
UHMW-PE	4.56	0.0336	2679.871
PEEK	97.92	0.047	5066.634

(Table 1-Material2)

Since the materials for this project are to be off-the-shelf components, all of these materials must be easily attainable. The final design calls for 0.5" diameter rods for the extension rods of the frame. Unfortunately the minimum diameter that UHMW-PE is supplied in is 0.75", which is too large for the extension rods due to the microphone mounting bases. Therefore the decision was made to use ABS for the three extension rods and UHMW-PE for the large diameter center main shaft. ABS is the second best material for the job based on its poor sound conduction characteristics. Although ABS could be used for the entire array, the outstanding properties of UHMW-PE cannot be ignored and using it as the main center shaft will play a great role in the overall vibration dampening of the acoustic array structure. The overall materials selected for the design are UHMW-PE for the main base, slider, and center main shaft; and ABS for the extension rods.

T-Base Array

Background

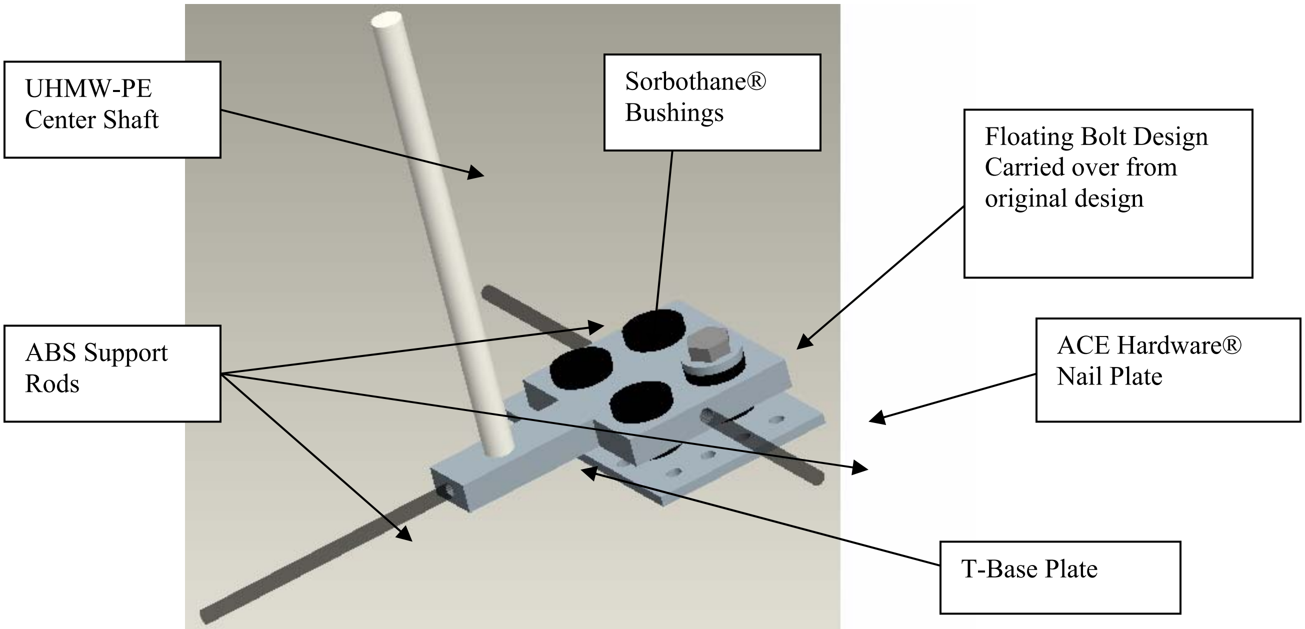
A project addition came about due to NASA stalling on the design parameters for the RDS robot in the spring semester. A smaller, half size version of the tetrahedral array was added to the spring requirements. The mini array is based of the same principles of the tetrahedral design, but inquires a different mounting setup and half the dimensions of the original version. The mini version mounting is to that of a VEX™ robot kit from Radio Shack®. This robot incorporates the same sensors as the NASA RDS robot and thus will serve the same purpose to allow for sensor integration between current and future systems.

Constraints

The mini array has the same constraints as the original tetrahedral array in that the geometry of the microphones must remain in a tetrahedron. The mini array is still based off of the 4 microphone algorithm, yet the spacing between the microphones was changed from 20 inches to 10 inches. Damping characteristics remain the same as the original tetrahedral array and the constraints are still for maximum damping of mechanical noise. Weight characteristics are at a heightened level due to the fact that the VEX™ robot must remain as lightweight as possible to allow for maneuverability and to reduce the tipping moment caused by the array. The mini array is attached to a standard nail plate from ACE Hardware® that was supplied from Eglin AFRL/MN. The mounting holes within the plate can not be modified and thus the mounting position for the mini array is fixed. The actuation constraint from the original frame has been cancelled for the mini array due to the relative complexity of integrating a controller system into the micro processing for the VEX™ robot. Overall, the final constraints for the mini array just entail half spacing for the microphones, mechanical noise damping, lightweight design, and a fixed frame with no actuation.

Design Process

Design of the mini array was determined to be based off of a T-base design type to allow for low cost and ease of manufacturing. The T-base serves as the vibration isolation plate, mounting structure to robot, and microphone support. The key concept is that the base is made from a single piece of UHMW-PE (Ultra High Molecular Weight Polyethylene). Using a single piece of material cuts down on material as well as possible vibration points within the structure. The overall design consists of just three main components: the T-base, center microphone shaft, and extension rods for the three base microphones. This simplicity of design allows for the same materials to be utilized from the original design: ABS rods, UHMW-PE base and UHMW-PE center shaft. Using the same materials allows for the same damping characteristics of the original array to carry over into the mini array. Ultimately, the mini array has similar characteristics in terms of material damping as the original array without the complexities of an actuation system causing more vibration points for mechanical noise to propagate. Figure T-base below gives a representation of the actual design characteristics of the actual mini array.



(Figure: T-Base)

Vibration Isolation on Tetrahedral Array

The vibration isolation for the acoustic array cannot be addressed at one single point in the frame. Since vibrations occur through very large frequency ranges a single device or mechanism to adequately damp these vibration impulses is inadequate. The solution is to use more than one mechanism throughout the frame. By utilizing multiple damping points the amount of vibration seen by the microphones can be substantially minimized.

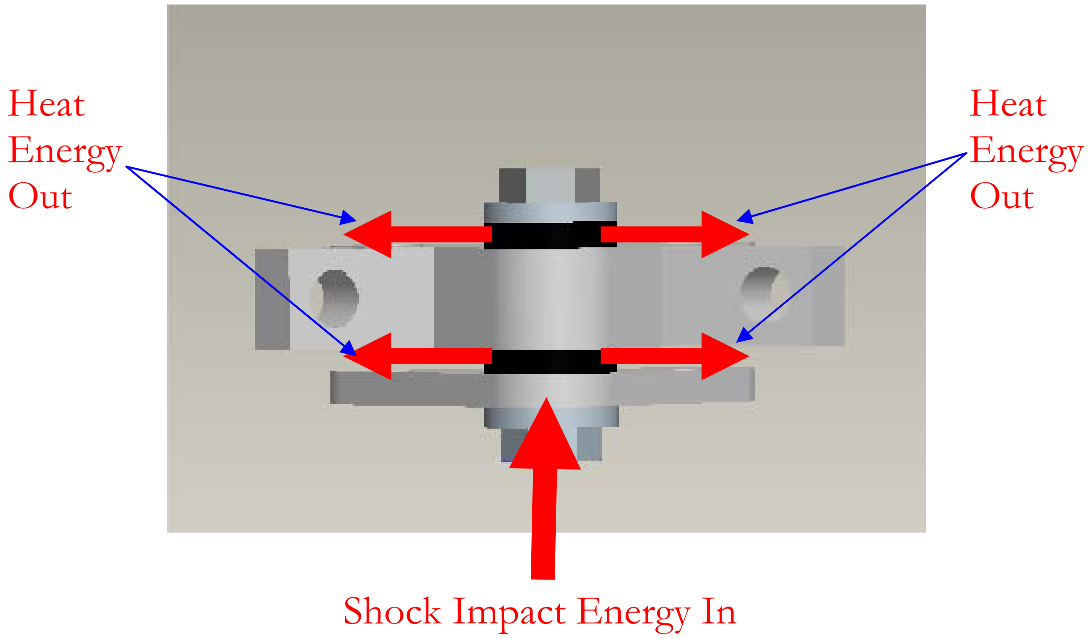
The tetrahedral four-bar slider frame will utilize three different vibration damping locations. The vibration points throughout the frame are through a floating bolt mechanism, throughout the array material, and a substrate under each microphone. Two main materials will be used for vibration damping, Sorbothane® and acoustic foam. Sorbothane® will be used in the floating bolt mechanism to physically decouple the array from the main structure. The acoustic foam will be used as a substrate between the frame and the microphone to physically decouple the microphone sensors from the array.

Vibration Isolation Materials

Sorbothane®

Sorbothane® is the primary material to be used for the vibration damping in this frame. Sorbothane® is a proprietary, visco-elastic polymer. It is a thermo set, polyether-based, polyurethane material (Sorbothane® Inc.). Visco-elastic means that it combines the superior damping characteristics of a viscous fluid and the shape holding characteristics of a solid.

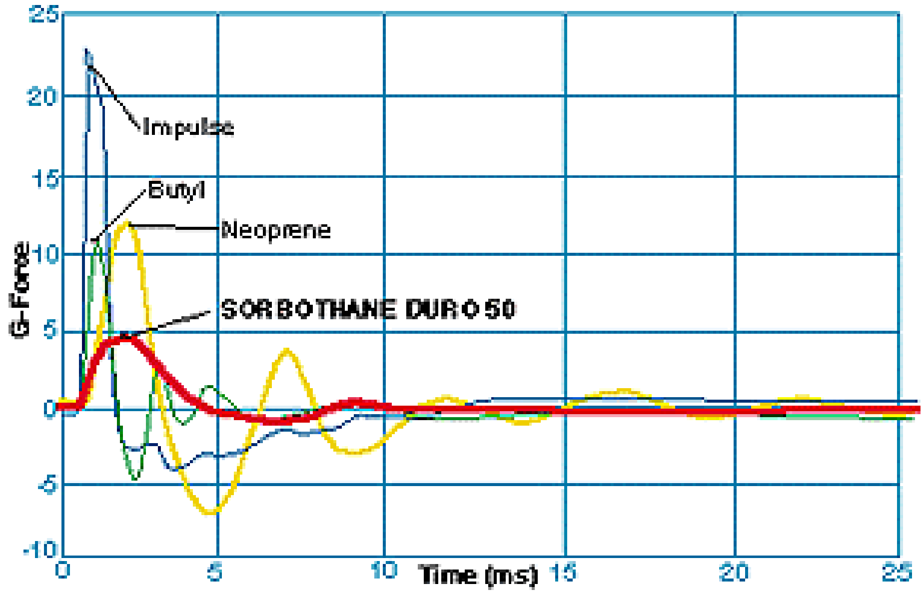
How this material works for vibration and shock damping is by turning mechanical energy into heat. Heat is generated by molecular friction as the material is deformed; the lost energy is called hysteresis. The impulse energy from the original source is translated perpendicularly away from the axis of incidence and its effect is pushed close to 90° out of phase from the original impulse (Sorbothane® Inc). See (figure 1-Impact) below for a depiction of how this process occurs when an impact in introduced into the system.



(Figure 1-Impact)

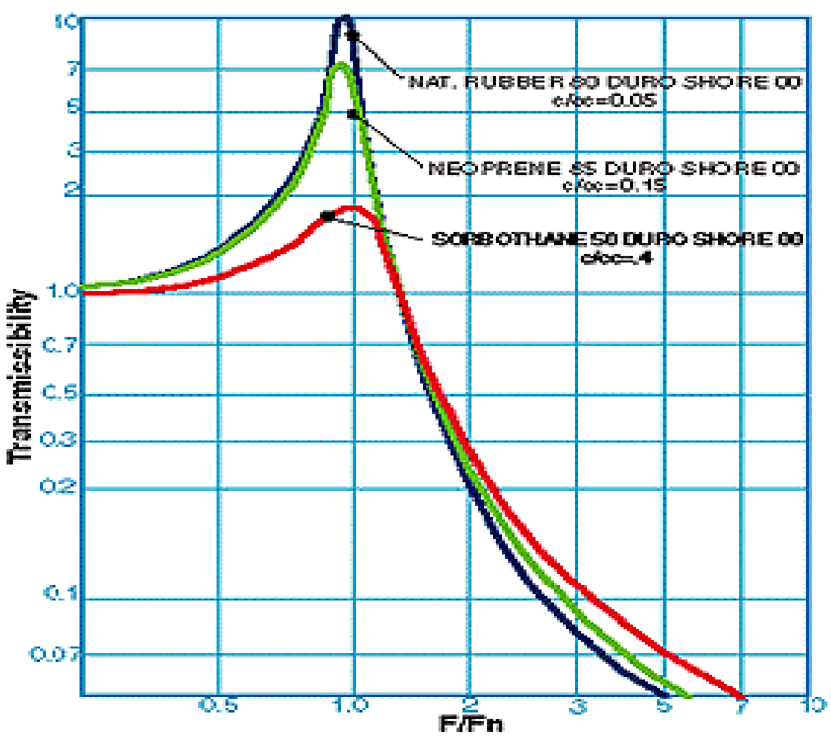
The high damping of polymers reduces impulse peaks of shock waves over a longer period of time. After an impulse, this material gradually and slowly brings the mass to rest (reference Sorbothane®). The impulse response of Sorbothane® compared to other materials can be seen in (Graph 1-Sorbothane). This material also exhibits low transmissibility at resonance. Isolation at large frequency ratios shows Sorbothane®'s ability to isolate vibration well. Depicted in (Graph 2-Sorbothane) is how Sorbothane® transmits vibration compared to other materials.

Sorbothane® Impulse Response (Sorbothane® Inc.)



(Graph 1-Sorbothane)

Transmissibility as a function of the Excitation Frequency/Natural Frequency Ratio (Sorbothane® Inc.)



(Graph 2-Sorbothane)

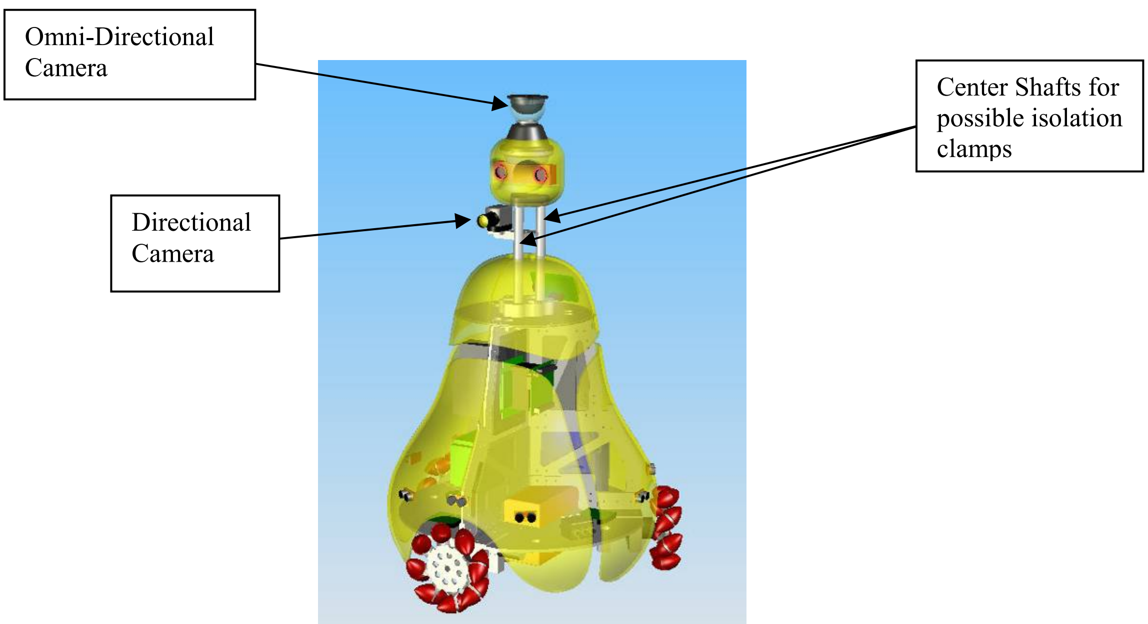
Acoustic Foam

Acoustic foam will be used as a substrate between the microphone and the tetrahedral frame. The foam will physically decouple the microphone board from any vibrations that might have made it through the other damping mechanisms as well as the frame itself.

Vibration Isolation Locations

Vibration Isolation at Robot Mounting Point

The array will be physically connected to the RDS robot by means of one or possibly two clamps bolted on the center shafts of the robot as in (figure 1-Robot). This is the starting point where the actual vibration inputs from the RDS robot will be conducted to the frame. This will be accomplished by insulating the clamp at the RDS robot-center-shaft contact point. By insulating that contact point with a layer of Sorbothane®, metal-to-metal contact of the robot and frame is entirely eliminated. Although not all vibrations will be completely eliminated at this point they will be greatly minimized.

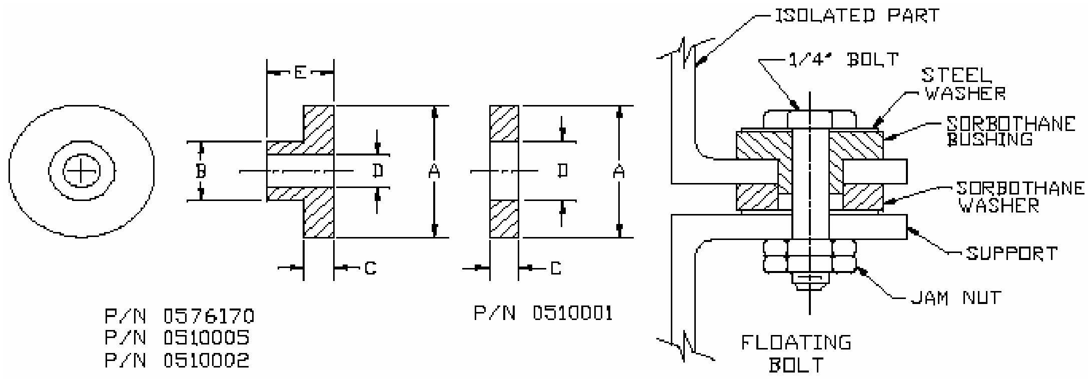


(Figure 1-Robot)

Vibration Isolation using a Floating Bolt

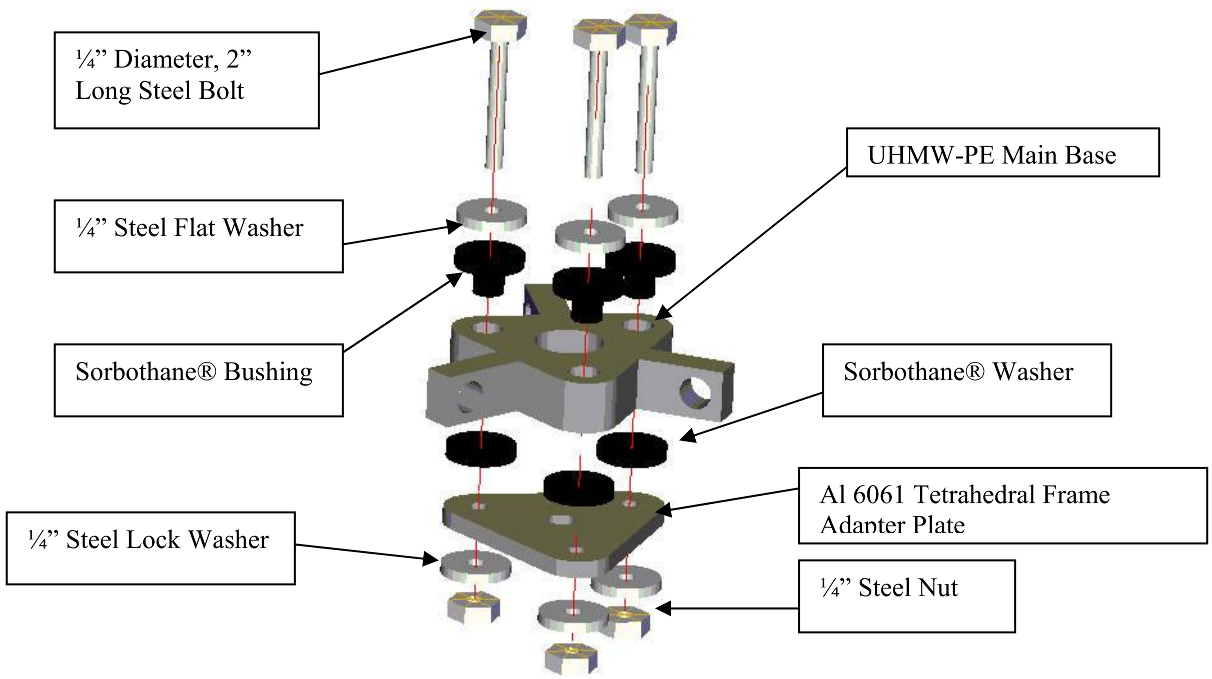
This will be the primary mode of vibration isolation since it maximizes the vibration damping characteristics of the Sorbothane®. By using the floating bolt design, the tetrahedral frame structure will be physically isolated from the robot at yet another point (array is already isolated from robot at the robot mounting point clamps). By applying a prescribed load (torque on the mounting bolts) on the Sorbothane® bushings and washers they will be able to damp out nearly all mechanical vibrations coming through that location. The following (figure 1-Bushing) is a drawing of the floating bolt design. Note: how the top part is physically isolated from the bottom part.

Floating Bolt (Sorbothane® Inc.)



(Figure 1-Bushing)

The following, (figure 2-Bushing) shows the actual isolation plate that will be used in the design.



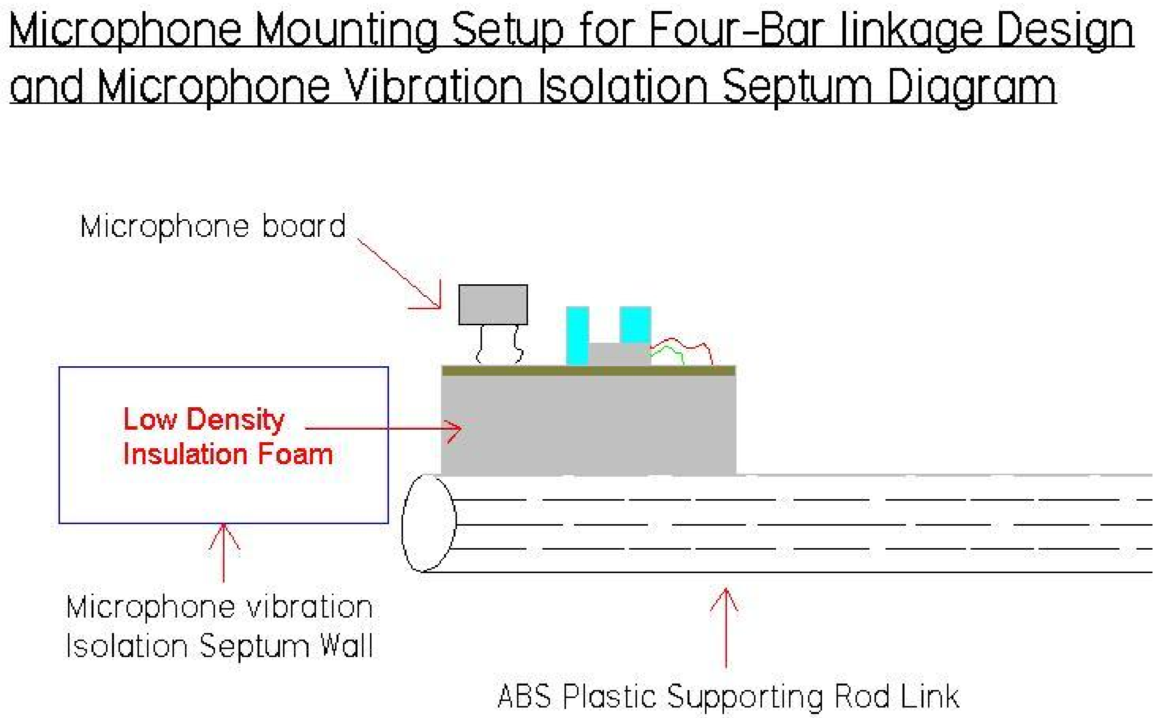
(Figure 2-Bushing)

Vibration Isolation through Array Material

This location was discussed thoroughly in the Four-Bar Tetrahedral Frame Material Selection section. (See previous section **Four-Bar Tetrahedral Frame Material selection)**

Vibration Isolation under Each Microphone (Septum Wall)

The last mechanism that will be used to damp the vibrations will be the Septum Wall. This wall consists of a substrate which will decouple the microphone board from the tetrahedral four-bar frame. The substrate will consist of a half inch thick section of low density foam. The microphone board will be mounted to the frame as seen in (figure 1-Septum).



(Figure 1-Septum)

Linear Actuation Device

Background

The secondary task for the completion of the design process was to automate the frame so that it would be collapsible to fold up to fit within the overall footprint of the RDS robot. The footprint was assumed to be 13 inches to underestimate the true size of the actual RDS robot that NASA Langley is producing. The reason for the discrepancy in this sense is that NASA Langley has yet to provide the dimensions of the actual robot that will be receiving the tetrahedral frame being designed. The design chosen for making this frame collapsible calls for some means of linear actuation to deploy and retract the microphone sensors. With this in mind, research was done to determine the best means to accomplish this motion.

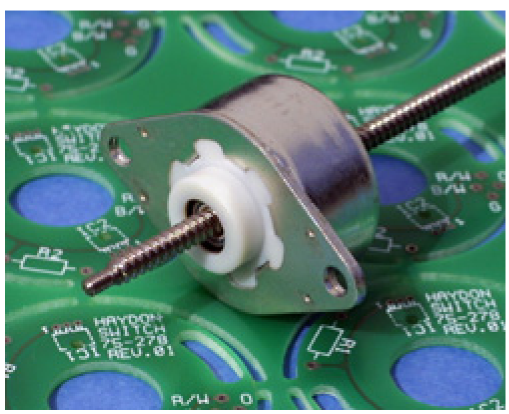
Findings

Calculations have shown that the actuator needed must be able to fit within the size of the four-bar tetrahedral frame and be able to lift at least a 10 Newton force with a 4 inch stroke. This calculation can be referenced in the calculations appendix along with other pertinent calculations to the overall design of the frame. With this force as a reference, research was done on linear actuators and solenoids that adhered to the constraints set forth by the design. The research turned up a company specializing in relatively small stepper motors that adhered to the design criteria. The company is Haydon Switch and Instrument, and they are located in Waterbury, Ct. and had all relative material required. Through conferencing with this company, the Z2684X-V stepper motor that they produce has the characteristics that meet the requirements needed. Some issues with this company are that they are also in the prototype phase with the longer lead screws required to attain a 4" stroke for actuation. Due to this step in the design only being a secondary task, the design has been updated to accept the stepper motor. The tetrahedral frame will be able to operate with or without this motor, making the design an evolving and robust one and can accept many modifications with minimal work. Dimensions and information on the Z2684X-V stepper motor may be found in the components appendix at the end of this document.

Stepper Motor Z-2684X-V

Stepper Motor Background and Operation

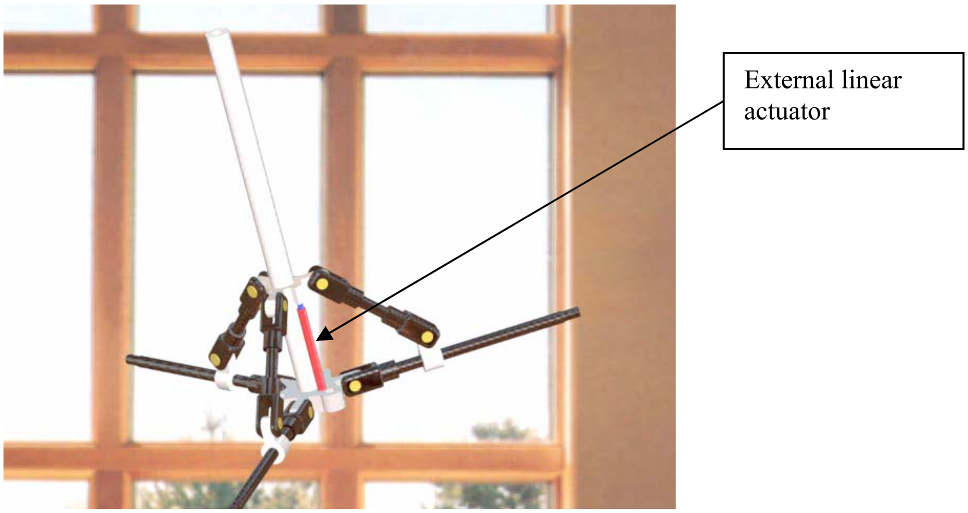
The actuation system chosen is a bipolar linear stepper motor from Haydon Switch and Instrument, Inc. and is there non-captive Z-2684X-V motor. The specifications for this motor can be found in the operations manual for the tetrahedral array housing. The operation of a stepper motor works on the principle of converting an electrical impulse into a mechanical movement. This method is accomplished by means of an electronic stator and an internal magnet rotor. An electronic controller produces a signal that becomes processed through the stator vanes in the motor assembly. The stator is comprised of a series of vanes which change in polarity as the signal is input. In the case of the bipolar stepper motor chosen for the project, a square signal is processed through the stator and causes a rotation of the center shaft which is linked to the central magnet. The magnet within the motor has two different polarities, north and south. As the stator vanes change in polarity, the corresponding magnet polarity is attracted and as the magnet is pulled toward the stator a rotation force is produced. This method is continued as the stator continually changes in polarity with the incoming signal. The speed of the motor is controlled directly by the incoming frequency and as the frequency is increased the motor speed is increased. The like is the case when the frequency is decreased, then the motor speed is decreased. The motor also is adjustable utilizing the current that passes through motor itself. By adjusting the current, the force that the motor produces may be increased or decreased. The motor does have a maximum current setting that must not be exceeded. The maximum current for the Z-2684X-V is 340 mA, and exceeding this limit will cause the motor to overheat. This motor has more than enough capabilities in terms of deploying and retracting the tetrahedral array housing.



(Z-2684X-V Bipolar Stepper Motor)
(Photo Courtesy HSI, Inc.)

Advantages/Disadvantages

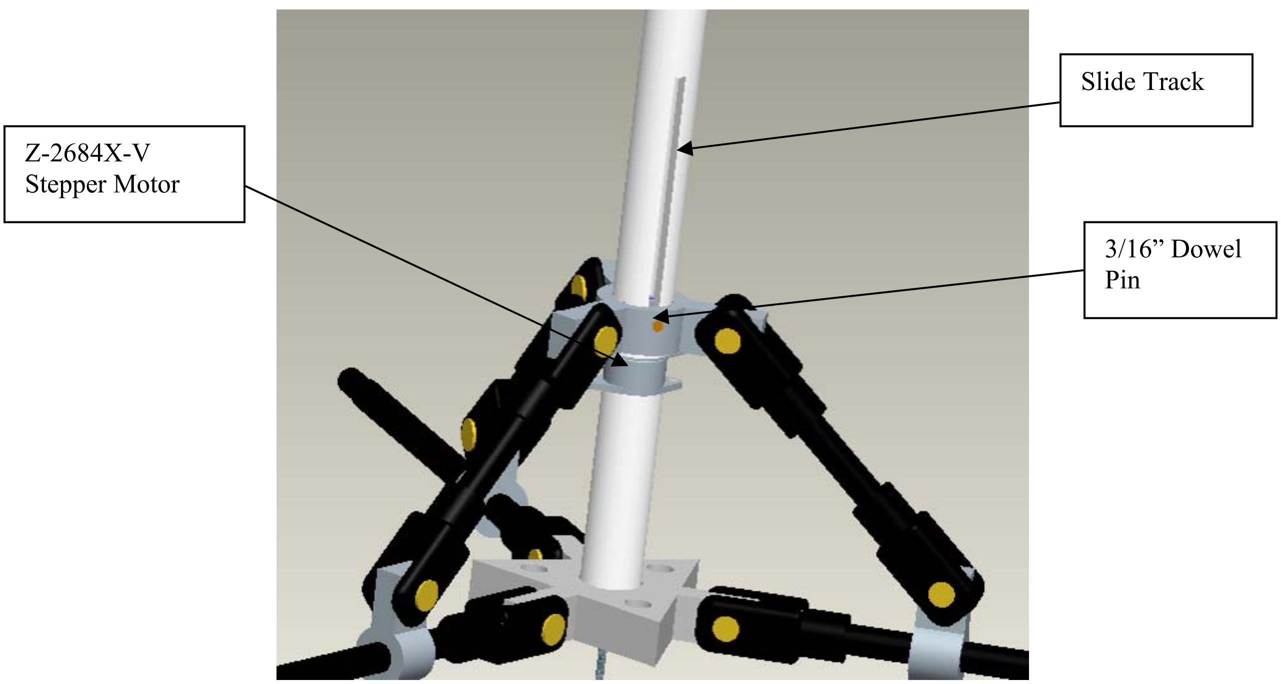
The bipolar stepper motor actuation method has many advantages when integrated into the tetrahedral array housing. These major advantages include computer or manual controllability and multiple adjustment methods. The Z-2684X-V can be controlled via a microprocessor encoded through a software program in a computer or by a manual stepper controller; this manual controller will be outlined further in this document. The computer controllability method has the advantage of making the stepper motor completely independent of human interaction for use in remote actuation as is the case with the integration to the NASA RDS robot. The micro-processing will be discussed in the future work and recommendations section of this document. The advantage of manual control of the motor via a controller board is that a person may directly adjust the potentials for the limits of the motor. This method also allows for demonstration purposes or in testing circumstances, as was the case for the tetrahedral array housing. Manual control of the motor allows for motor control without the use or need of computer software or programming. The advantages of the different adjustment methods are outlined according to the type of adjustment available for the motor. The motor has three major adjustment possibilities: current, frequency, and voltage. The advantage to current adjustment is such that the motors output force to the lead screw can be increased or decreased. This is especially desirable when the motor is being placed under large loads where a high current is required. In the case of the tetrahedral array, a low current setting is required to match the force of the motor to the load imposed by the array. By adjusting the frequency, the time and speed at which the motor is actuated may be set to the user's needs. This advantage allows for the tetrahedral array to be deployed or retracted at different rates depending on what the user requires at any given time. Voltage is also adjustable for the motor and this is desirable due to the power source that the RDS robot is utilizing. The RDS robot is using a 5 volt supply and the motor is adjustable down to 5 volts. Obviously, there is the advantage of using the motor in different applications depending on what power supply is used as the motor is adjustable up to 12 volts. There is however one inherent disadvantage to using the stepper motor in the tetrahedral array housing. The motor has a temperature rise of 75ºC or 135ºF and it is made of stamped steel. The array itself is made up of UHMW-PE which is a form of a polymer and has a possibility of melting if the motor is overheated for any reason. Given these advantages and disadvantages, the Z-2684X-V motor is the preferred choice for actuation of the tetrahedral array housing.



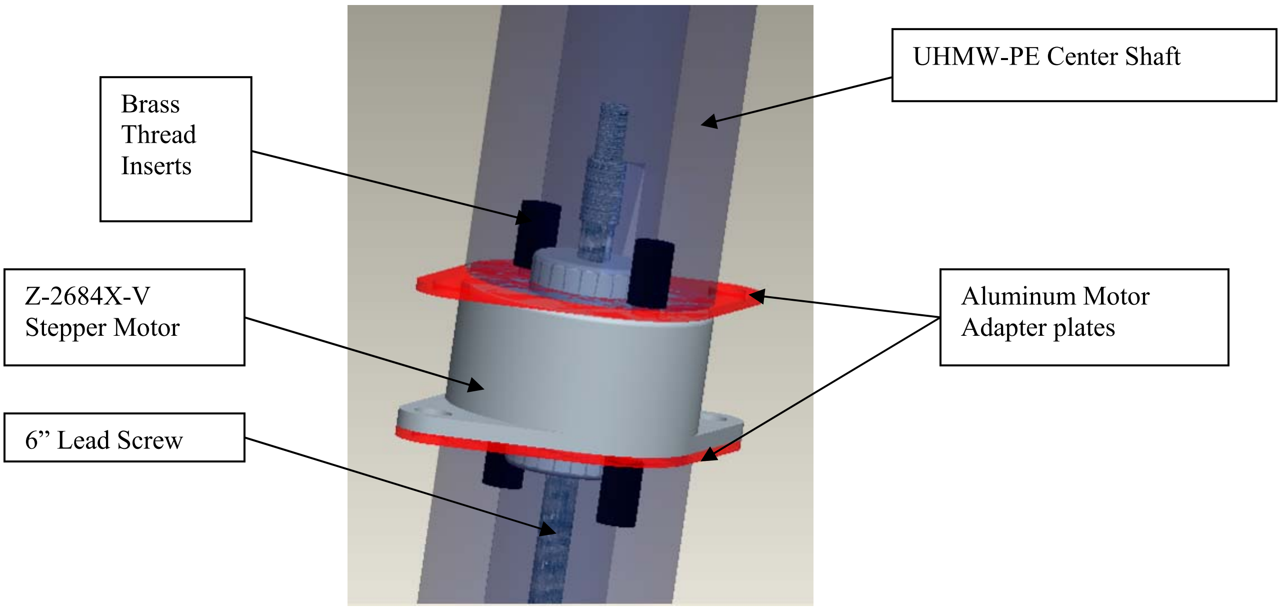
(Linear Actuator External Actuation)

Integration into Design

The original actuation method chosen for the tetrahedral array housing was that of an external linear actuator system. This method of actuation can be seen in the above figure, (linear actuator external actuation). External actuation presented too many complications with part availability as well as design complexity. For these reasons, the Z-2684X-V stepper motor was chosen and integrated into the tetrahedral array housing. Integration of this motor presented its own complications, but overall became a much better and straightforward method of actuation. Adapter plates had to be integrated to hold the motor centered over the center axis of the UHMW-PE center shaft. These adapter plates can be seen in the below figure, (adapter plate integration). By placing the motor over the center axis of the center shaft, the original external moment created by the external linear actuator has been eliminated. Integration of the stepper motor also creates a dual slide track slider mechanism. This is accomplished via the internal surface of the center shaft and the external surface of the center shaft. Dual sliders keep the lead screw constantly centered and provide more surface area to resist binding from the actuation process, this can be viewed in the below figure (dual slide track). The benefits of this design create a smoother transition throughout the actuation process than would have been seen if an external actuator was used. Overall, integration of the Z-2684X-V stepper motor provides the tetrahedral array housing with a much sleeker design and a better method of actuation throughout the entire range of motion.



(Dual Slide Track)



(Adapter Plate Integration)

Conclusion

Actuation of the tetrahedral array housing is accomplished via the Z-2684X-V stepper motor. This motor allows for a wide range of adjustability methods such as current, frequency, and voltage. These adjustments allow for changing of the force, time and speed, and power supply that the motor can output or handle. Utilizing total adjustability, the motor can operate at multiple settings based upon the needs of the system at the time. The integration of the dual slider slide track method allows for a smooth transition from the array’s fully retracted position to the fully deployed position and vice versa. Implementation of the Z-2684X-V stepper motor into the tetrahedral stepper motor allows for total design control and integration into the complete system operations.

Components/Parts Selection

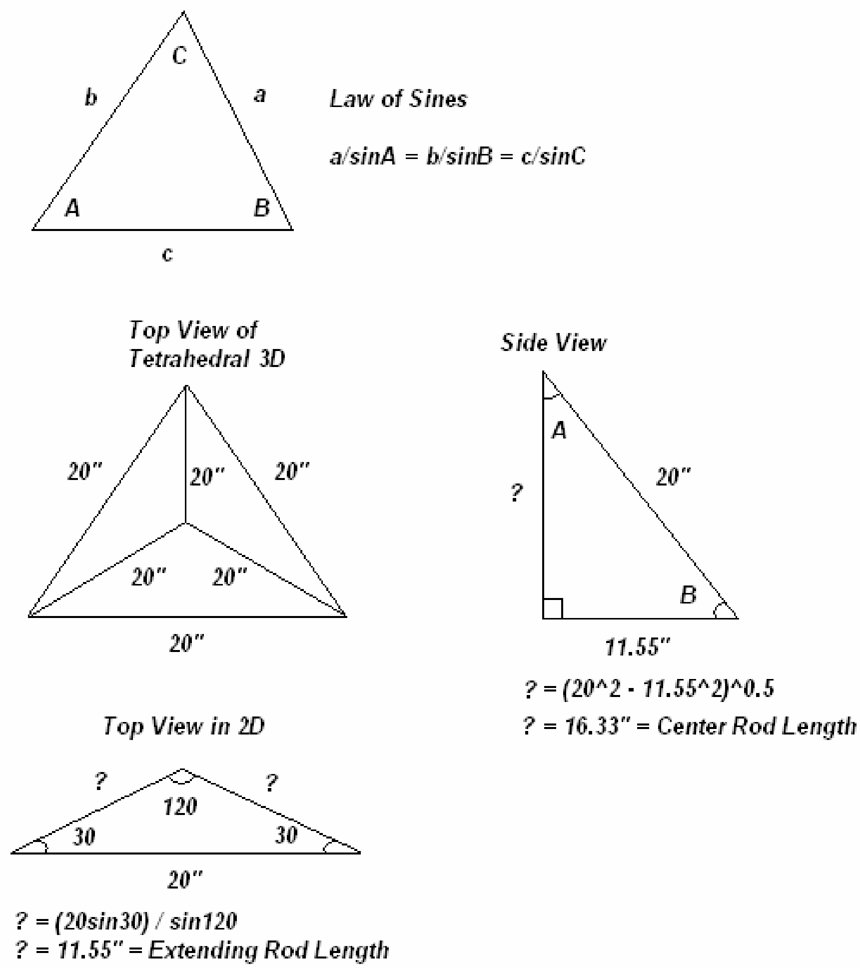
Specific parts are required for the assembly of the tetrahedral array frame as well as the T-base array and include small fasteners and clevis joints. Specific components were researched that held characteristics that adhered to the constraints of using off-the-shelf, lightweight components for a mechanical vibration damped housing. The fasteners that hold the frame to the robot were not of real concern in transferring mechanical vibration to the frame due to the floating bolt design chosen to suspend the frame from the actual robot. With this being the case, standard steel, ¼" bolts of 2" length were chosen for the tetrahedral frame and shorter 1" length bolts were used for the T-base array. The cylindrical joints poised a much different issue in adhering to the constraints set forth by the project objectives for the tetrahedral array. Any time a joint is used on the frame, an inherent introduction of mechanical noise through friction and tolerances is produced. The type of joint chosen that best fit the design constraints was a standard clevis joint. The clevis joint allows for adjustability as well as clearance between the extension rods and the main base of the tetrahedral frame. These characteristics allowed for the least amount of material as possible for the main base and extension rods. The issues facing the overall selection of the specific clevis joint used came down to weight, tolerance, and mechanical vibration transmission. The clevis joint chosen was an Igubal® clevis joint. This specific joint was chosen due to the characteristics that it possesses: lightweight, high tensile strength, vibration dampening, noise dampening, tight tolerance, and adjustable. These clevis joints are made from igumid G, which is a lightweight material that is suitable for the design. The components chosen for this design ensure that the costs and availability are not an issue and that all components are off-the-shelf, standard components.

Pertinent Calculations

Pertinent engineering calculations were required to ensure that the design chosen would conform to all the necessary parameters set forth in the (**Constraints**) section. The calculations done were for material usage, material savings over other designs, speed of sound through a material, material and component costs, actuator force required, and pendulum swing momentum.

Material Usage Calculation

The overall material usage calculation centered on minimizing the amount of total material required by a specific design. This ensures that the cheapest possible frame could be developed. The fixed parameter for this calculation was the 20 inch center-to-center distance for the four separate microphones and the tetrahedral configuration. With this in mind, the sine law was implemented to determine that the four-bar tetrahedral slider frame did have the optimized amount of material usage over the other designs. See (figure 1-Calc1) for the material rod length calculations and see the calculations appendix for the full calculations for all the designs considered.



(Figure 1-Calc1)

Material Savings over other designs

The material usage calculations proved that the four-bar tetrahedral slider design used the least amount of material possible. From this calculation, the percent savings over choosing the four-bar design over the other designs was calculated. This calculation was done to determine just how much more effective the four-bar design was in terms of material required. The results were pretty profound in that the four-bar required 34% less material than the other designs, see (figure 1-Calc2). To see the entire calculation reference the calculations appendix at the end of this document. This is due in part to the fact that all the designs other than the four-bar required the same amount of materials for production.

Percent Savings in Material

$$\%\_Savings := 1 - \frac{Total}{Total\_O}$$

$\%\_Savings = 34.07\,\%$

(Figure 1-Calc2)

Speed of Sound Conduction through materials

The speed of sound through the various materials researched was required to determine which material had the lowest sound conduction speed. The materials that this value was calculated for were polycarbonate, ABS, Carbon, Graphite, UHMW-PE, PEEK (Polyetheretherketone). The material with the slowest conduction speed would be the material with the best vibration mitigation characteristics. From the calculations done using the equation in (figure 1-Calc3), two materials warranted themselves feasible for the four-bar slider frame design. UHMW-PE (Ultra High Molecular Weight Polyethylene) had the lowest sound conduction at 2679 ft/s, while ABS (Acrylanitrile Butadiene Styrene) came in second with a conduction speed of 4254 ft/s. Both of these materials were determined to be used in the design due to restricted availability of UHMW-PE, otherwise UHMW-PE would have been used for the entire frame. For a more detailed depiction of these calculations see the calculations appendix.

The speed of sound in a solid

$$\nu = \left(\frac{E}{\rho}\right)^{\frac{1}{2}}$$

Where E is Young's modulus and rho is the density of the matrial.

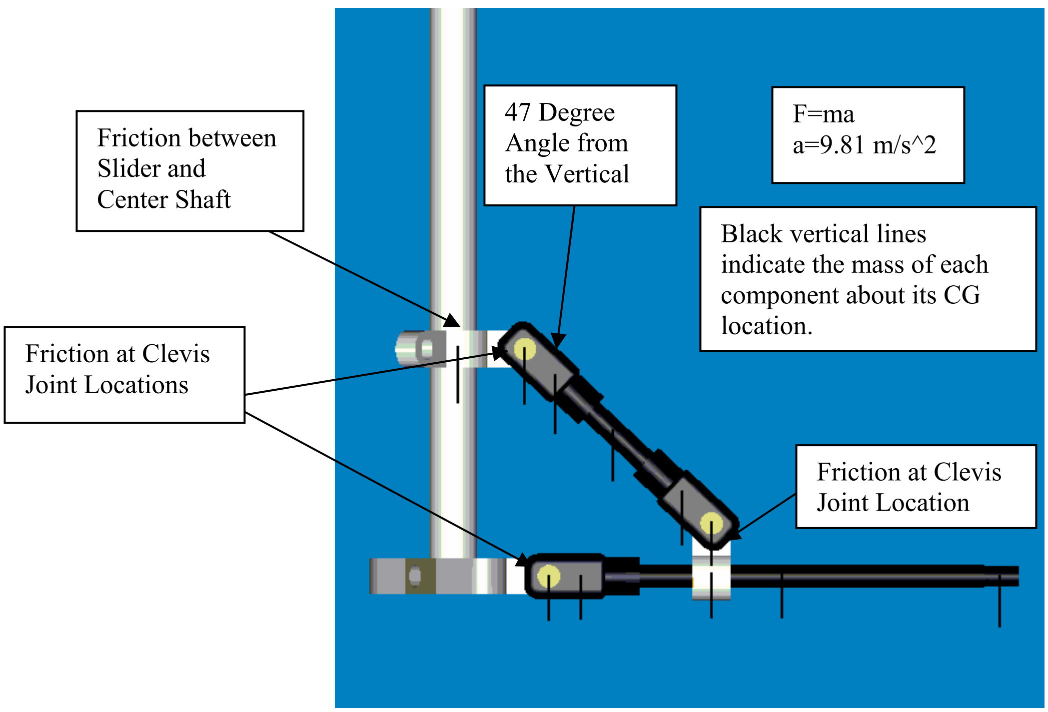
(Figure 1-Calc3)

Material and Component Cost Calculation

The constraints from Eglin AFRL/MN included designing for a cost effective design with an emphasis on having the cheapest design possible and still meet all the constraints. The cost calculation was a simple process of sourcing the materials chosen for their cost and determining the amount of materials and components required to achieve the specific design from the ideation phase. The most cost effective design found from these calculations came out as the four-bar slider frame design with a cost of approximately $134. The actual build cost for this design and its build process are outlined in the (**Cost Analysis**) section of this document. To see the actual calculation and the values for the other designs, reference the calculations section in the appendix.

Actuator Force Required Calculation

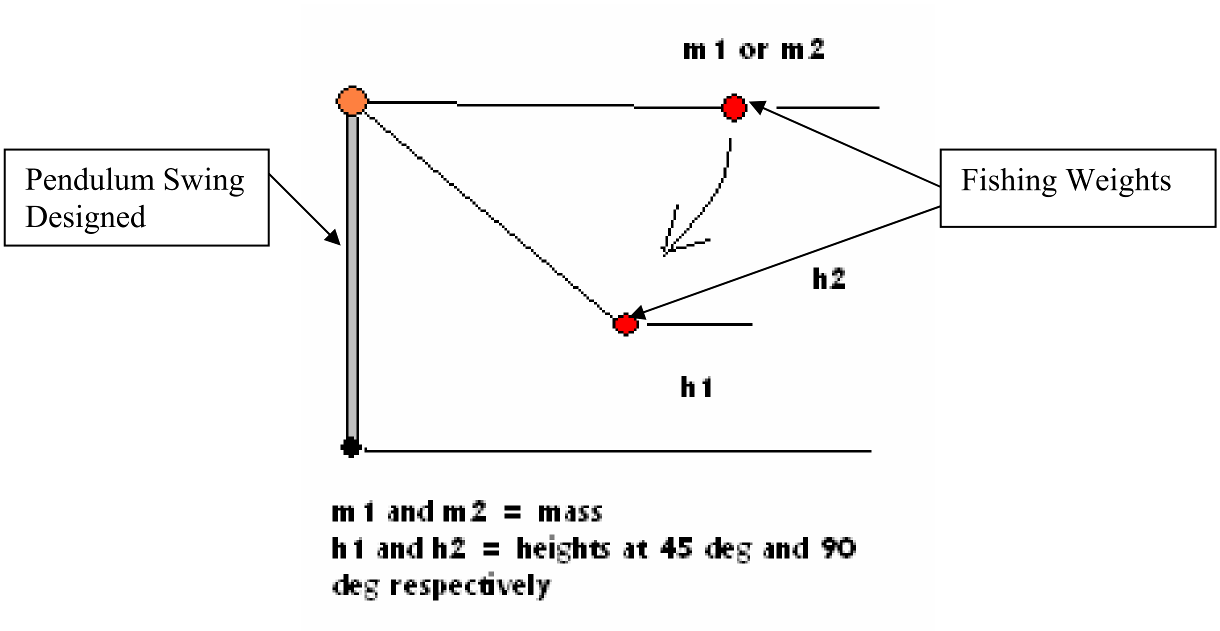
The design of the four-bar tetrahedral slider frame included the implementation of some type of autonomous linear motion. After some research, it was determined that the best means for such motion would be a solenoid or linear actuator. The actuator force required had to be calculated to determine which of the two devices would be better suited at deploying and retracting the four-bar slider mechanism on the frame. This calculation was done by utilizing the basic force equation seen in (figure 1-Calc4). Friction between the joints as well as the slider to the center shaft also had to be taken into account. Utilizing the force equation and introducing the force of friction that needed to be overcome, the actuator force required was determined to be approximately 10 Newton's. The free body diagram in (figure 1-Calc4) depicts how the actuator force required was calculated. The detailed calculation can be seen in the calculations section of the appendix.



(Figure 1-Calc4)

Pendulum Swing Momentum Calculation

The testing phase of the design process of the four-bar frame required an impact test to generate vibration within the structure of the frame. To generate this type of vibration a pendulum swing was built to produce an impact as the weight impacts the extension rod at the bottom of the arc. The momentum of this weight needed to be varied to produce different frequencies of vibration to simulate different types of possible impacts. To determine the momentum that the weights would be impacting the rods at, a simple momentum calculation was done. The free body diagram in (figure 1-Calc5) depicts how the momentum was calculated. To determine momentum, the conservation of energy equation was used to determine the velocity of the weights being dropped at different heights. The momentum equation was then used to determine the momentum that the weights would be impacting the extension rods with. The momentum of the weight can be seen in (table 1-Calc5) below and a detailed calculation may be viewed in calculations appendix at the end of this document.



(Figure 1-Calc5)

Mass	Starting Angle	Velocity at Bottom of Arc	Momentum
0.125 lb	90º	10.356 ft/s	**1.295 lb*ft/s**
0.25 lb	90º	10.356 ft/s	**2.589 lb*ft/s**
0.125 lb	45º	5.606 ft/s	**0.701 lb*ft/s**
0.25 lb	45º	5.606 ft/s	**1.401 lb*ft/s**

(Table 1-Calc5)

Cost Analysis

Background

The cost analysis for the project has been calculated based upon the components required to assemble the test frame as well as the final four-bar tetrahedral slider frame. Various companies have been contacted and sourced to determine the availability of the components that are required to complete the design process. The main components were outlined above in the component selection section as well as the material selection section. The constraint to use as many off-the-shelf components as possible led to the research to find suppliers that could deliver the components required for design completion in a timely manner. Companies sourced were McMaster Carr®, Lowe's®, Igus® Inc., and Sorbothane® Inc. McMaster Carr® was sourced namely for the availability of the UHMW-PE and the ABS polymers. Lowe's® was sourced for the mounting and securing hardware. Igus® Inc. provided the clevis joint assembly information. Sorbothane® Inc. provided the information for the vibration bushings and washers. The constraint to keep the cost to a minimum from the AFRL/MN is outlined below in the form of price calculations for components required to complete the design.

Parts Required

The analysis up to this point has not considered the components necessary to attach the frame to the robot. The overall system costs for the tetrahedral frame at this juncture are outlined below with part numbers and suppliers listed.

Part	**Supplier**	**Part #**	**Price**
UHMW-PE Hollow Rod 0.99” OD x 0.45” ID, 2 ft long	McMaster Carr®	8705-K332	$4.56/Foot x 2
ABS Rod 0.5”OD, 5 ft long	McMaster Carr®	8587-K43	$1.46/Foot x 5
UHMW-PE Disk 5” OD, 3” Long	McMaster Carr®	9352-K21	$10.72 Each
Aluminum Plate 6061, ¼” Thick, 6” x 6” Square	McMaster Carr®	3511T11	$41.31 Each
Igubal Clevis Joint	Igus® Inc.	GELIK-07	$9.57 Each x 9
¼” Steel Bolts, 2” Long, ¼”-20 Thread	Lowe’s®	136012	$2.30 Package
¼” Steel Nuts, ¼”-20 Thread	Lowe’s®	136006	$1.04 Package
¼” Flat Steel Washers	Lowe’s®	136002	$1.04 Package
¼” Lock Washers	Lowe’s®	135999	$1.04 Package
Set Screws, ¼”-20 Thread, 0.3875” Long	Lowe’s®	137266	$0.68 Package x 2
Sorbothane® Bushings	Sorbothane® Inc.	0510001	$1.08 Each x 3
Sorbothane® Washers	Sorbothane® Inc.	0510002	$1.08 Each x 3
Total			**$167.84**

Design Cost Conclusion

The overall component costs come out to be approximately $170.00 to build both the test frame and the final tetrahedral frame not including a stepper motor. This is possible by utilizing components from the test frame in the final frame. This greatly reduces the cost by not having to buy more materials to build the final test frame. The cost analysis did not include factoring the costs for shipping or taxes. These parameters cannot be included at this time due to the fact that this value is dependent upon location of the order and destination of the shipping. This cost analysis falls well within range of the original cost estimates outlined in the final design selection report. All necessary components, including the integration of the stepper motor into the tetrahedral frame provides a total final cost of approximately $200 for the prototype tetrahedral array and $30 for the T-base array. These costs are based on the fact that many of the parts are based on bulk ordering and making a prototype costs more than mass production of the these arrays.

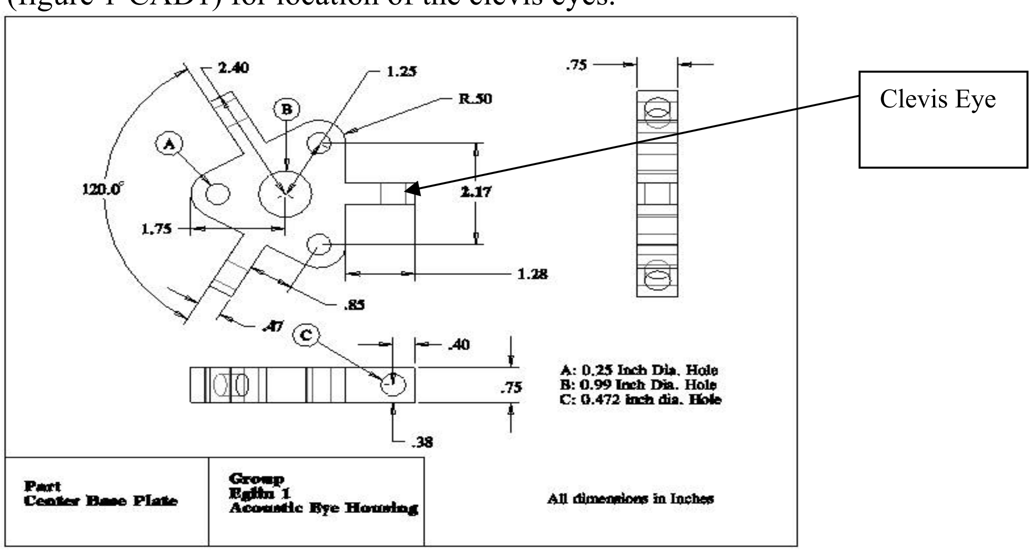
Build Procedure Four-Bar Tetrahedral Frame

Purpose

The building process chosen for the tetrahedral four-bar slider frame will be that of a bottom up assembly. This type of assembly will ensure that the frame is assembled without having extraneous steps or having to continually change the orientation of the frame to assemble each successive part. The purpose of this means of assembly process is to reduce the number of complex steps as well as overall complexity in the assembly phase.

Machinability

Machinability has profound effects on the overall design of the tetrahedral four-bar slider frame because the frame had to be designed around machining parts rather than casting or molding due to cost. The design of the main base of the frame is utilizing UHMW-PE which is an extremely machinable polymer. The design of the main base itself had to be such that it is practical in a machining sense while conforming to the constraints of the overall design. With this in mind, the design of the main base includes features such as equidistant straight cuts, through holes of equal diameter, and clevis eyes with 0.01" tolerances. The design of the slider piece entails using UHMW-PE so that the same material is used throughout the structure as well as to cut down on overall cost. The design features that are attractive in a machinability sense for the slider are that the overall finish will not affect the performance of the sliding action. The overall machining time for the entire frame is minimized by the fact that only two parts need to be machined for the assembly process. The CAD drawing in (figure 1-CAD1) depicts the dimensioning that the main base plate needs to have in order for the tetrahedral design to be effective. The tolerance that is of main concern for the machined components is only that of the clevis eye protrusions and this value must not deviate more than 0.01" in either direction, see (figure 1-CAD1) for location of the clevis eyes.



(Figure 1-CAD1)

Parts List

1) One, UHMW-PE Disk, 5” diameter by 3” length.
2) One, UHMW-PE Rod, 0.99” diameter by 36” length.
3) One, ABS Plastic Rod, 0.5” diameter by 60” length.
4) Nine, Igubal® Clevis Rod End Joint Assemblies (Includes pin and clip).
5) Three, Standard Steel Set Screw, ¼” diameter by 3/16” long with ¼”-20 thread.

Assembly (Four-Bar Tetrahedral Array Frame)

The assembly process for the frame is accomplished keeping in mind the least amount of steps possible to complete the assembly. With this in mind, the bottom up assembly process is outlined as follows for the tetrahedral four-bar slider frame.

Preparation

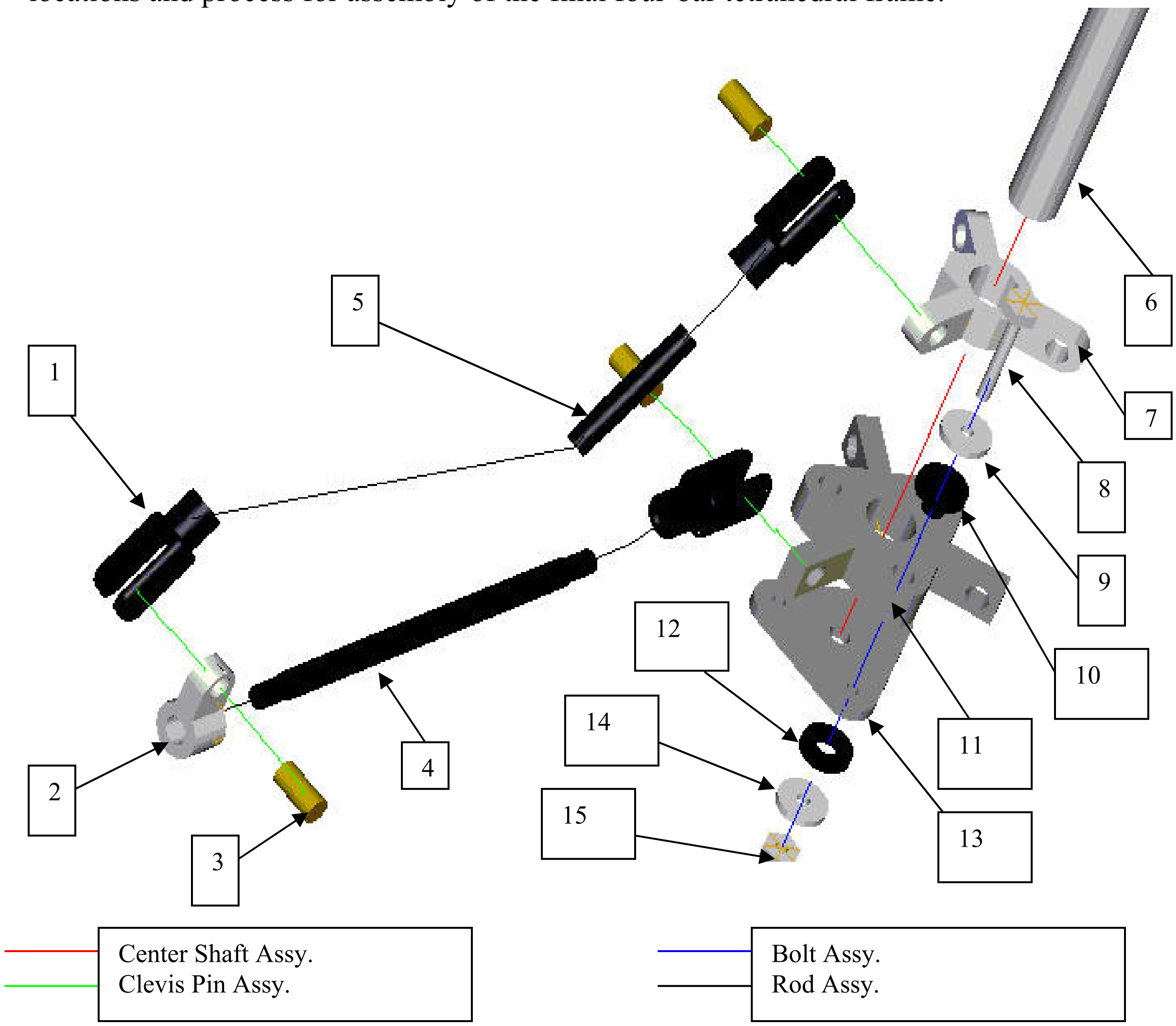
1) Machine the main base, slider, and 90º rod links out of UHMW-PE according to CAD drawings in the CAD appendix.
2) Cut ABS rods for the extension arms to length specified in CAD drawing and thread one end to specifications in the CAD appendix.
3) Cut ABS rods for the connecting arms to length specified in CAD drawings and thread both ends to specifications.
4) Cut UHMW-PE rod for the center main shaft to the length specified in CAD drawing.

Actual Assembly

1) Place main base on press and press fit center shaft into through hole in the center of the main base. (Note: bottom of center shaft must be flush with bottom of main base).
2) Slide slider over the center main shaft and let rest on the main base for the time being.
3) Thread the threaded end of the 3 extension rods into the clevis rod ends. (Note: extension rod lengths are to be adjusted at the end of the assembly process).
4) Slide the three 90º rod links over the open end of the 3 respective extension rods and secure with set screw to the extension rod in desired location. (Note: 90º rod links are adjusted at the end of assembly process).
5) Place the clevis joint end of the extension rod assemblies over the clevis eye protrusions in the main base and secure with clevis pin and clip.
6) Thread clevis rod ends into both sides of connecting arms.
7) Place clevis joint end into clevis eyes on 90º rod links on respective extension rod arms and secure with clevis pins and clips.
8) Slide slider up from main base approximately half way up center main shaft to aid in the assembly of the connecting rods in the next step.
9) Take the other side of the connecting arm assemblies and attach clevis joints into clevis eyes on slider and secure with clevis pins and clips.
10) Attach microphone base plates to the ends of the three extension rods as well as the top of the center main shaft.
11) Insert the three Sorbothane® bushings into the mount holes on the top side of the main base plate part.
12) Place ¼” flat washers over the Sorbothane® bushings.

13) Insert ¼" bolts through the flat washers and the bushings.
14) From the bottom up, place the three Sorbothane® washers over the protruding ends of the ¼" bolts.
15) Slide robot adapter mount plate over the three protruding ¼" bolts over top of Sorbothane® washers.
16) Slide three lock washers over the protruding ¼" bolts and let rest on the robot adapter mount plate.
17) Place ¼" nuts over protruding ¼" bolts and secure in place.

Reference the exploded state drawing in the (figure 1-CAD2) to determine the locations and process for assembly of the final four-bar tetrahedral frame.



(Figure 1-CAD2)

Bill of Materials for (Figure 1-CAD2)

1) UHMW-PE Clevis Joint
2) UHMW-PE 90º Rod Link
3) Igubal® Clevis Pin
4) ABS Extension Rod
5) ABS Connecting Arm
6) UHMW-PE Center Main Shaft
7) UHMW-PE Slider
8) ¼" Bolt
9) ¼" Flat Washer
10) Sorbothane® Bushing
11) UHMW-PE Main Base Plate
12) Sorbothane® Washer
13) ¼" Aluminum Plate
14) ¼" Flat Washer
15) ¼" Nut

Tetrahedral Test Frame

Purpose/Description

A test frame was built for the purposes of experimentation of the materials selected for the tetrahedral acoustic sensor housing. This frame is of the non-folding type as mentioned in prior documentation and entails phase 1 of the design process. The purpose of the experimentation is to obtain data that can be used as a baseline to the actual final folding housing. The actual test processes used are explained in the next section and cover the specifics behind how the data being obtained through the test frame were analyzed.

Build List

A number of components are required to build the actual test frame seen in (figure 1-Frame). The actual parts list is as follows for the assembly from the bottom up.

1) 3, 7.5 Volt DC motors
2) 3, ¼" Diameter x 9/8" Long x 9/4" Across, U-Bolt Assemblies
3) 3, ¼" Diameter by ¼" Deep, Nylon Spacers
4) Acrylic Main Base, Equilateral Triangle with 4" Length sides by 1" Deep
5) 3, 20" Long by ½" ABS Diameter Extension Rods
6) Acrylic Microphone base, 1.75" Wide x 1.75" Long x 1" Deep
7) UHMW-PE Center Shaft, 16.33" Long x 1" Diameter
8) 3, ¼" Bolt Assemblies with 3 Lock Washers, 3 Flat Washers, and 3 Nuts.

(Figure 1-Frame)

Build Procedure

The test frame itself is a solid structure configured in the required tetrahedral array. The microphone mounting locations are 20 inches from center to center of each microphone mounted at each apex of the tetrahedron frame. The test frame consists of 3 ABS extension rods, UHMW-PE center shaft, acrylic microphone and main base plates, and corresponding hardware. An acrylic main base was machined into an equilateral triangle with sides of 4 inches and an overall depth of 0.5 inches. Holes, 0.5 inches in diameter, were machined into each side of the triangle and the 3 ABS extension rods of 20-inch length were then inserted. Set screws were used to secure the extension rods into the main base. A 1-inch diameter through hole was then machined into the main face of the base plate for the UHMW-PE center shaft of 16.33 inches of length to be press fit into place. Finally, 3 0.25 inch through holes were machined into the main face of the base plate for mounting bolts. Microphone mount plates were made next and utilized acrylic squares with 4, 1/8 inch through holes drill and tapped for the microphone board to be mounted in place. The dimensions of these mount plates were 1.75 inches by 1.75 inches with an overall depth of 1 inch. A 0.5-inch through hole was machined longitudinally through the mount plates such that they could be press fit over the extension rods at the apexes of the tetrahedron. The robot mock up base was the final piece to the test frame and was made from ¼-inch thick aluminum that was 4 inches by 8 inches. Corresponding ¼-inch through holes were machined through the main face such that the acrylic main plate could be bolted to it. For further testing purposes, 7.5 Volts DC motors were mounted to the aluminum plate to mimic the DC servomotors on the actual robot that the tetrahedral housing would be mounted. The motors will be run using 6 Volts to get a realistic portrayal due to the actual robots being powered by a 6-volt source.

Analysis

The actual tests performed on the housing were a motor vibration test, rod impact test, and test frame impact test. These are outlined here only as how they reference to the test frame. The actual testing processes are outlined in the next section. All tests were performed with the aforementioned test frame to get viable data from the experiment. The motor vibration test consisted of powering one DC motor with a 6-volt power supply and acquiring the vibration that travels through the material of the test frame to the microphone mounted at the apex of the tetrahedron. The rod impact test was done using weights of various masses to simulate the extension rods of the test frame impacting objects as the robot maneuvers. The test frame impact test was done to obtain data on the types of vibrations that the test frame would transmit to the microphone sensor as if one of the actual mecanum wheels were dragging as the robot maneuvers. The purposes of these specific types of tests are related to the determined vibrations that the actual robot will produce. These specific tests are a viable representation of the actual vibrations that the robot will produce in the real-world environment.

Test Procedures

Background

The testing on the tetrahedral test frame will be done to simulate the different types of mechanical vibrations that the actual robot will generate and transmit to the microphone sensor. To accomplish this task, three separate experiments were generated based upon the three most prominent means of mechanical vibration generated by the actual robot. The three separate vibrations are DC motor vibrations, frame impact vibration, and vibration from a direct impact to the robot structure or the tetrahedral array itself. Accurately portraying these different vibrations requires utilization of the exact materials used in the actual robot as well as the final tetrahedral array frame. By doing this, the data from the experiments will provide a very good approximation as to the actual vibrations that the actual robot will produce. The reasoning for the discrepancy is the fact that we are unable to attain the exact robot that NASA is building and also a prototype robot can not be built because we have not been supplied with the final specifications of the actual robot. Despite these hurdles, a mock robot base was built using an aluminum base with three DC motors attached in the same triangular configuration as the actual NASA robot. The mock base is a scaled down version of the actual profile for cost reasons for testing purposes only and this will have negligible affects on the overall testing. These same testing procedures were adapted to the actual four-bar tetrahedral frame that will be proposed to AFRL/MN at the end of the spring term.

Experimental Tests

The experimental testing of the microphone and tetrahedral array is outlined in the following sections in the form an experimental lab handout. The purpose of using this format is such that the tests are reproducible and outlined such that any person can use this document to conduct an exact replication of the experiment.

Experiment 1: DC Motor Vibration Test

Objective

The purpose of this experiment is to analyze the mechanical vibration transmitted by a DC motor as voltage is metered between 3 and 6 volts. The analysis will be focused around the transmission of direct and damped mechanical vibration through a tetrahedral test frame as if the actual DC servomotors on the actual robot were being throttled for navigation purposes. The experiment is to determine the actual mechanical vibration through the ABS extension rods as well as the UHMW-PE center shaft. The second part of this experiment will address a damped mechanical vibration by utilizing an acoustic foam substrate.

Apparatus

The components necessary to conduct the experiment are listed as follows.

6-Volt (Variable to 3 Volt Minimum) DC Power Supply

This will be used to simulate the actual power supply that the NASA robot will use. This device will power 1 DC motor as well as the microphone sensor.

7.5 Volt DC Radio Controlled Car Motor

This motor will be used to generate the mechanical vibration that will be analyzed. Even though this motor is rated to 7.5 Volts DC, it will be run at 6 Volts max and throttle to lower settings. This is to simulate the actual power settings that the actual NASA robot will utilize.

Tetrahedral Test Frame

This is the actual mode for mechanical vibration transfer between the simulated robot base and the microphone sensor. This frame acts as the test bed for the material vibration properties.

Simulated Robot Base

This will allow for the mounting of the DC motor as well as the mounting of the tetrahedral test frame.

Microphone Sensor

This is the actual sensor being mounted on the actual robot. This device will be used to acquire the mechanical vibration from the tetrahedral test frame. The microphone sensor is mounted at the apexes of the tetrahedron.

LabVIEW Software and Computer

This is the actual means of transferring the analog data from the microphone sensor to the computer for digital analysis.

Multimeter

A multimeter will be used to measure the actual voltage of the power being supplied to the motor as well as the microphone sensor.

Two Cardboard Boxes

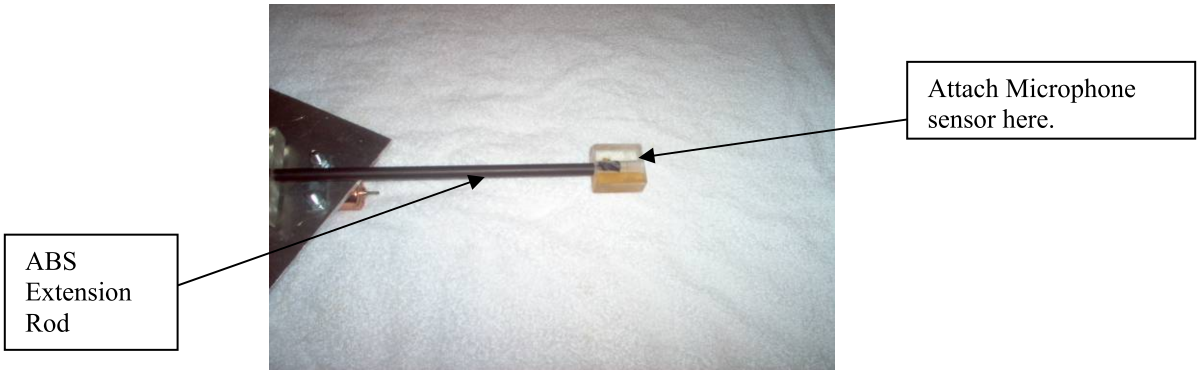
These boxes will be used to act as non-acoustic environments. The boxes will serve to separate the microphone sensor from acoustic sounds from the DC motor. They also allow only mechanical vibration to be transmitted to the microphone sensor from the DC motor.

Experimental Procedure

Note: The tetrahedral test frame and mock robot base were pre-assembled and bolted to one another.

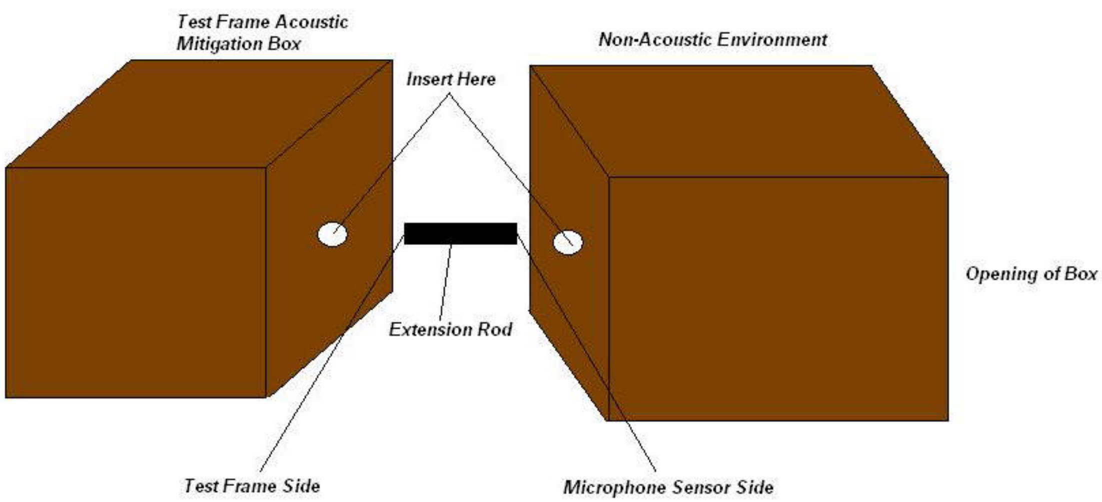
Part I.) Direct Mechanical Vibration through tetrahedral test frame.

1) Attach microphone sensor to mounting point on tetrahedral test frame using double-sided (non-padded) tape. Reference (figure 1-EXP1).



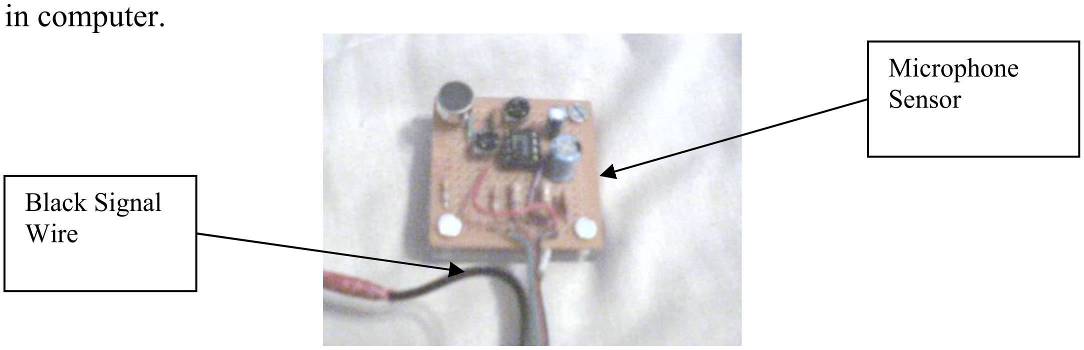
(Figure 1-EXP1)

2) Place extension rod into hole in one cardboard box and through other cardboard box hole as seen in (figure 2-EXP1).



(Figure 2-EXP1)

3) Connect 6-Volt power supply to DC motor and microphone sensor.
4) Connect black signal wire from microphone sensor in (figure 3-EXP1) to A/D board in computer.



(Figure 3-EXP1)

5) Connect multimeter to power supply and record the actual power supply voltage.
6) Using LabVIEW, switch on the DC motor at 6-volts and begin data acquisition over 20 second time interval. (This is all accomplished inside the LabVIEW program).
7) Repeat step 4, for 3-volts and 4.5-volts to simulate the throttling of the servomotors on the actual NASA robot for control purposes.

Part II. Mechanical Vibration Damped by a substrate

1) Place acoustic foam between microphone sensor and mounting point on tetrahedral test frame.
2) Repeat steps 3-7 from Part I above.

Conclusion

After all data is acquired, generate dynamic response curves from the voltage output versus the time interval. Finally, compare the direct vibration data to the substrate-damped data.

Experiment 2 and 3: Impact Tests (Extension Rod and Robot Base)

Objective

The purpose of this experiment is to determine the transmission of mechanical vibration to the microphone sensor as if an extension rod or robot base were being impacted by an external force. The key to this experiment is to impact the extension rod or robot base at different locations with different forces. In doing so, a range of different mechanical vibration responses will be generated. This test will allow for the generation of material characteristics of ABS plastic in response to a dynamic impulse.

Apparatus

The components necessary to conduct the experiment are listed as follows.

6-Volt (Variable to 3 Volt Minimum) DC Power Supply

This will be used to simulate the actual power supply that the NASA robot will use. This device will power the microphone sensor.

Tetrahedral Test Frame

This is the actual mode for mechanical vibration transfer between the simulated robot base and the microphone sensor. This frame acts as the test bed for the impact vibration test.

Microphone Sensor

This is the actual sensor being mounted on the actual robot. This device will be used to acquire the mechanical vibration from the tetrahedral test frame. The microphone sensor is mounted at the apexes of the tetrahedron.

LabVIEW Software and Computer

This is the actual means of transferring the analog data from the microphone sensor to the computer for digital analysis.

90 Degree Pendulum Swing

This is used for the physical dropping of the weights to generate the impact force on the ABS extension rod.

Fishing Weights: 2 and 4 Ounce

These weights will be used to generate the force necessary to impact the ABS extension rod.

Protractor

A protractor will be used in order to set the angle of the fishing weights.

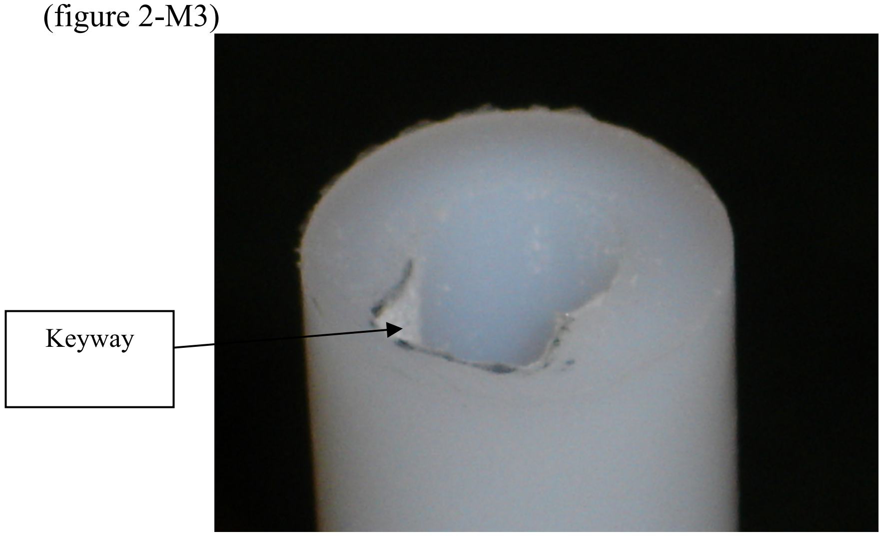
Experimental Procedure

1) Attach microphone sensor to mounting point on tetrahedral test frame using double-sided (non-padded) tape. Reference (figure 1-EXP1).
2) Connect 6-Volt power supply to microphone sensor.

3) Connect black signal wire from microphone sensor in (figure 3-EXP1) to A/D board in computer.
4) Connect multimeter to power supply and record the actual power supply voltage.
5) Insert pendulum swing into center shaft as seen in (figure 1-EXP3).

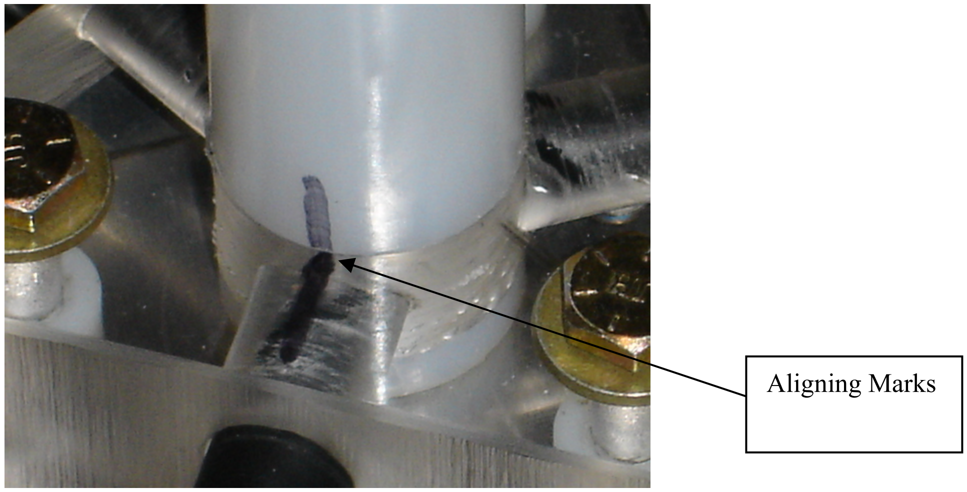
(Figure 1-EXP3)

6) Align metal L-bracket in pendulum swing with keyway in center shaft as seen in (figure 2-M3)



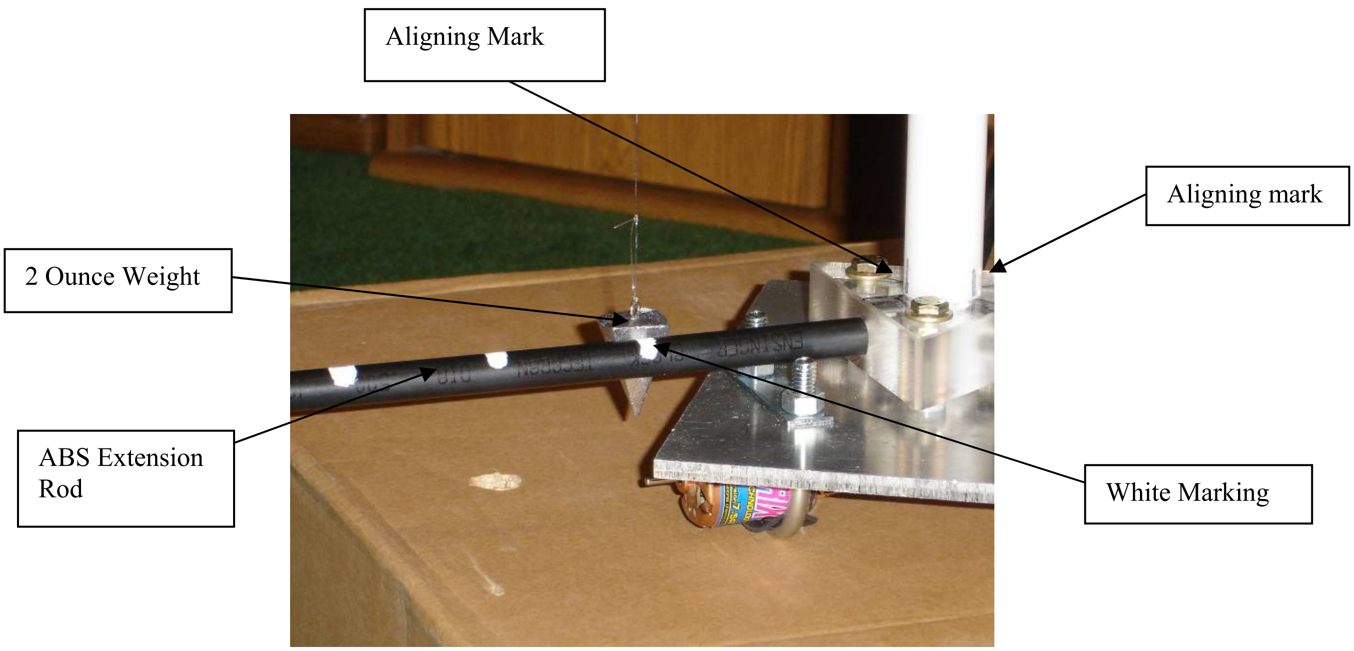
(Figure 2-EXP3)

7) Align center shaft markings with test frame acrylic base markings as seen in (figure 3-EXP3).



(Figure 3-EXP3)

8) Place 2 ounce fishing weight on pendulum swing using fishing line.
9) Align center of gravity of 2 ounce fishing weight to strike center line of ABS extension rod.
10) Adjust 2 ounce weight to first white marking closest to center shaft on ABS extension as shown in (figure 4-EXP3).



(Figure 4-EXP3)

11) Hold weight perpendicular to center shaft and at 90 degree angle to crossbar.
12) Simultaneously release weight and begin data acquisition in LabVIEW as in (figure 5-EXP3).

(Figure 5-EXP3)

13) Repeat step 11 using a 45 degree angle.
14) Repeat steps 10-13, aligning the weight with the next closest marking to center shaft each time until reaching the last marking.
15) Repeat steps 8-14, using 4 ounce fishing weight.

<u>Conclusions</u>

After testing is complete, review impact data to ensure that the values generated are realistic and viable. Review impact test for possible errors, which may include inconsistencies in impact location on extension rod, and note them to ensure that comparison testing conducted later during the project will be as accurate as possible. A substantial margin of error is expected during testing due to design limitations.

Testing

Characterized Test Parameters

Specific parameters had to be characterized to be able to analyze how well the system was at damping out possible vibration that will be impinging upon the system. The most important parameters for characterizing the effectiveness of the array design were decided to be the critical damping coefficient, impulse magnitude, and settling time. These parameters will be used to gauge whether or not there is significance to the methods used for damping out the possible system vibrations.

Critical Damping Coefficient

The critical damping coefficient is the rate at which an impulse shock wave will attenuate back to an equilibrium value. The significance of this parameter in the vibration testing is that it will gauge how well the vibration control techniques are damping as compared to the original non-damped system.

Impulse Magnitude

The impulse magnitude is the physical strength that an impact shock will impose upon the system. This value is significant to the array frame due to the fact that it will show how well the damping techniques are able to mitigate the impulse response. Decreasing the impulse magnitude will mean that the system is physical resisting the impulse by converting the excess energy into heat as it is damped out of the array.

Settling Time

The settling time is an important parameter in that it characterizes the time over which the system is affected by an impulse are vibration wave. The shorter the settling time, the better the system is at damping out the impulse or vibration wave.

Tests Run

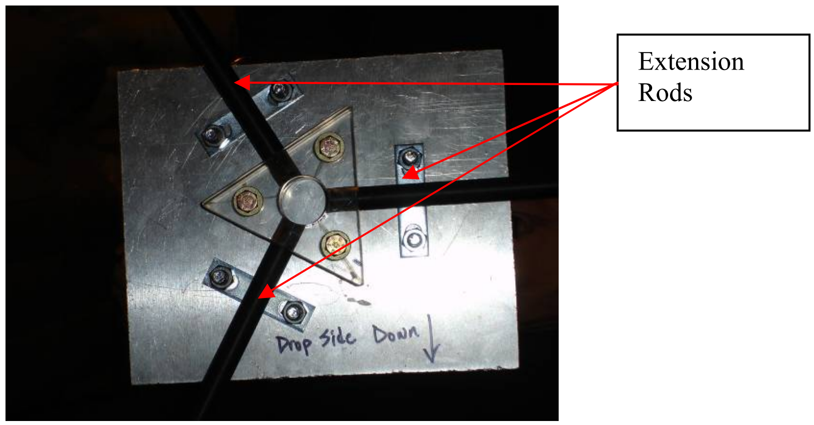
Three tests were run in order to characterize the different methods by which vibrations would normally be likely to impact the array system. The three tests used were a base impact test, rod impact test, and a DC motor test. These tests served as a physical simulation of the actual real world navigation state that the RDS robot is likely to see on a normal operation basis.

Base Impact Test

The base impact test consisted of using an aluminum test base that mimics the aluminum frame of the RDS robot. This test reveals the amount of propagation of the incoming mechanical vibration source to the microphone sensors themselves. This will serve as the most important test because the RDS robot will be continuously sending mechanical vibrations through its structure and this test will be able to quantify how that will affect the microphone sensors.

Rod Impact Test

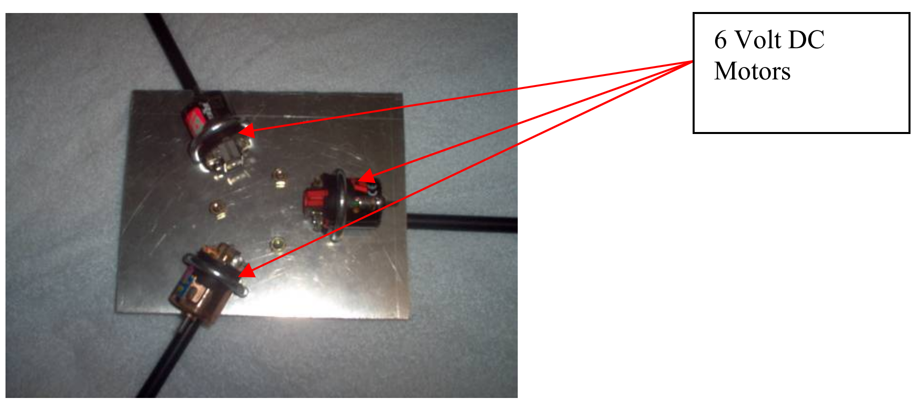
The rod impact testing serves as a means to characterize the result of the RDS accidentally impacting an obstruction with one of the tetrahedral array's extension rods in the deployed position as seen in (figure Ext-Base) below. This test will characterize the result of direct shock impulse propagation through the extension rod to the microphone sensor.



(Figure Ext-Base)

DC Motor Test

The DC motor test simulates the three DC motors that are incorporated into the RDS robot design. These motors produce significant mechanical vibration caused in part by motor operation and wheel slip across a surface. To mimic this effect, 3, 6 Volt DC motors were incorporated into the test base as seen in (figure Motor-Base) below. The parameter of interest in this test is the excitation of the microphone sensor as well as the impulse magnitude of the response from the mechanical vibration to the sensor.



(Figure Motor-Base)

Test Data Analysis

Data Acquisition

The primary means of data collection was through the use of LabVIEW software. LabVIEW is computer-based software that implements an internal I/O card in the computer for data acquisition. The data acquisition card or DAQ card has input pins that receive voltage from the sensor that is attached to it. The LabVIEW interface is configurable and adjustable depending upon which type of data sampling is required. For the purposes of acquiring data from the microphone sensor, only one configuration was utilized within the LabVIEW interface. Voltage input to the DAQ card was the only sampling that was required to characterize the arrays performance. In the LabVIEW interface there are adjustments for number of scans, scan rate, and upper and lower voltage limits. The number of scans is the total amount of sample size that is required for a specific test. The number of scans was standardized for all tests and was 21,000 scans. The scan rate is the number of scans per second that the DAQ card will receive. For standardization, all the tests implemented 7,000 scans per second. The voltage adjustment was set up such that the limits of the microphone sensor would not be exceeded. The limits for the microphone sensor were 6 to -6 volts. This process and interface was utilized for all tests and provides uniformity as well as simplicity to the testing process.

Data Analysis Methods

Several techniques were utilized to analyze the data that was acquired using the LabVIEW software interface. Microsoft® Excel was used as the spreadsheet analysis method to convert the LabVIEW data into quantifiable numbers that could be analyzed. Data from LabVIEW was saved to an Excel file format such that the Excel program could interpret the data that the LabVIEW software was collecting. After the data was converted it was analyzed using basic graphing techniques of y versus x to obtain the impulse response from the testing. Where y was the voltage output from the DAQ card from the microphone sensor and x was the time over which the data was collected. Once the graphs were generated, a process of using pixilation was utilized to determine the period of the half sine wave that was generated during the impulse testing.

Pixilation

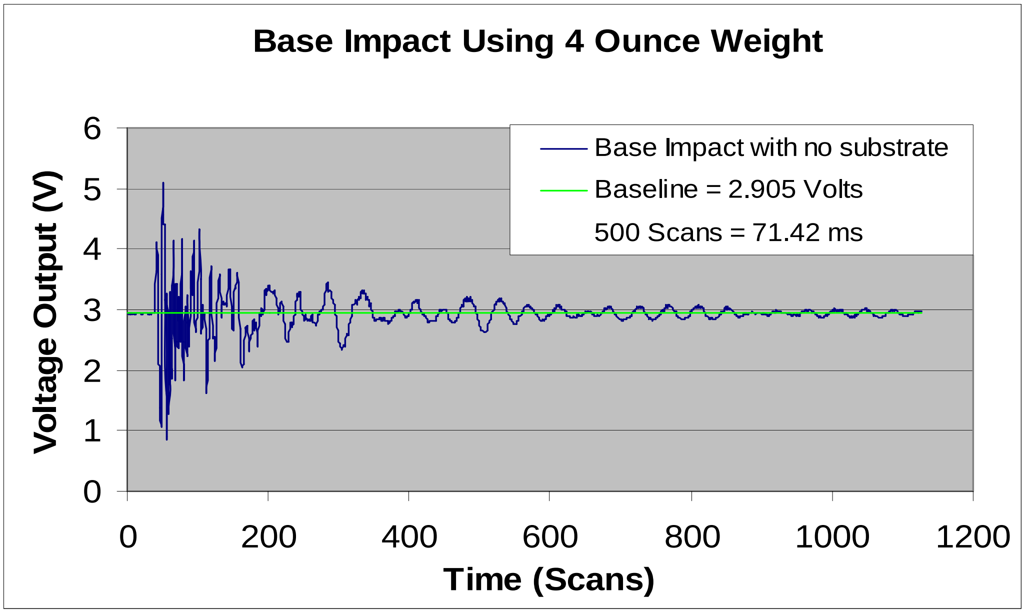
Pixilation is the process of using a computer program such as Microsoft® Paint™ to analyze graphs. This process entails determining the location of pixels with respect to an origin to determine the quantifiable data points within the graph. The process used for the analysis of the test data was such that the graphs generated in Excel were imported into Paint™ and analyzed. Once the graphs were in Paint™, the pixel coordinates for the origins of each graph were determined. The graphs imported were of half sine wave type and the peak to peak coordinates were obtained to calculate the period by which each graph depicted. This pixilation process was used thoroughly throughout the data analysis phase of the project.

Preliminary Testing

Preliminary testing was required to characterize the vibration waves that were being experienced by the microphone sensors. This data had not been previously ascertained and was required to serve as a comparison platform between the damping effects implemented upon the system versus the non-damped system case. Testing consisted of the aforementioned DC motor test, rod impact test, and base impact test. The characteristic parameters under scrutiny were the critical damping coefficient, impulse magnitude, and settling time. Preliminary testing was done by using the tetrahedral test frame and hard mounting the microphone sensor directly to the extension rod, as in (figure 1-PTest) below to be able to collect the full magnitude of the impulse response.

(Figure 1-PTest)

Base Impact Test



(Figure Impact_Base)

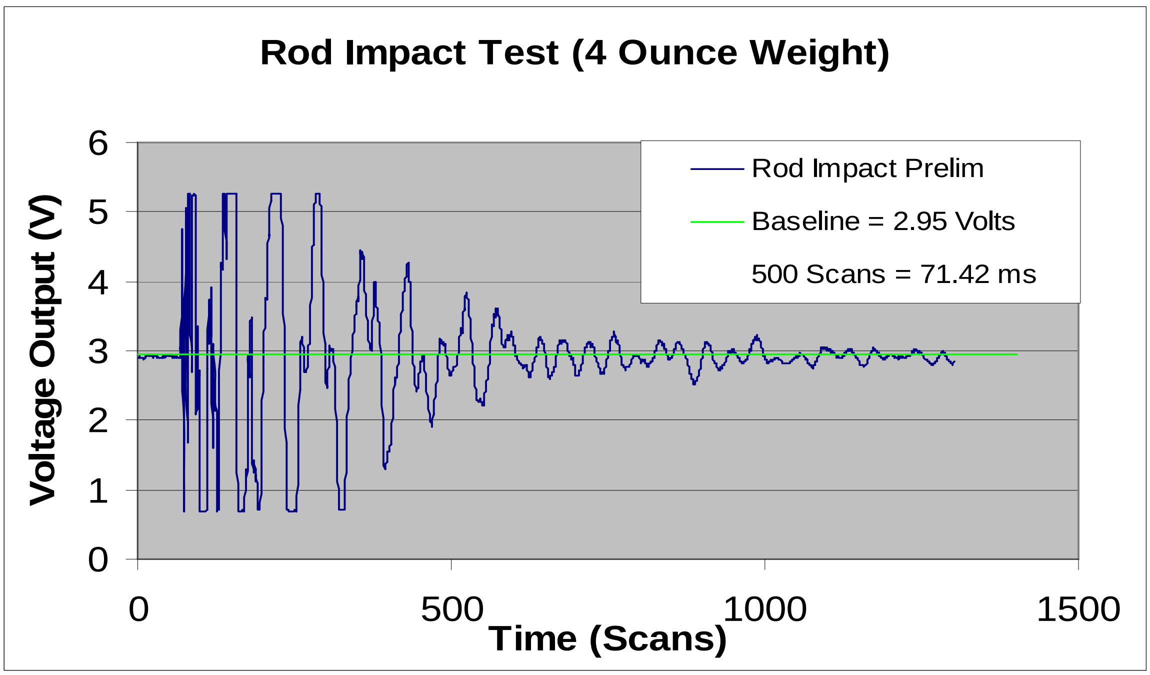
Base Impact Characterization

The base impact test was accomplished by using a 4 ounce weight to impact the simulated RDS aluminum test base. This impact caused the half-sine wave depicted in (figure Impact_Base) above. Characterization of this data provides the base values for the critical damping coefficient, impulse magnitude, and settling time for the system. The preliminary values for the base impact test can be seen in (table Impact_Base) below.

Base Impact Characterization Results		
Critical Damping Coefficient	Impulse Magnitude	Settling Time
866.5 kg/s	5.073 Volts	161 ms

(Table Impact_Base)

Rod Impact Test



(Figure Impact_Rod)

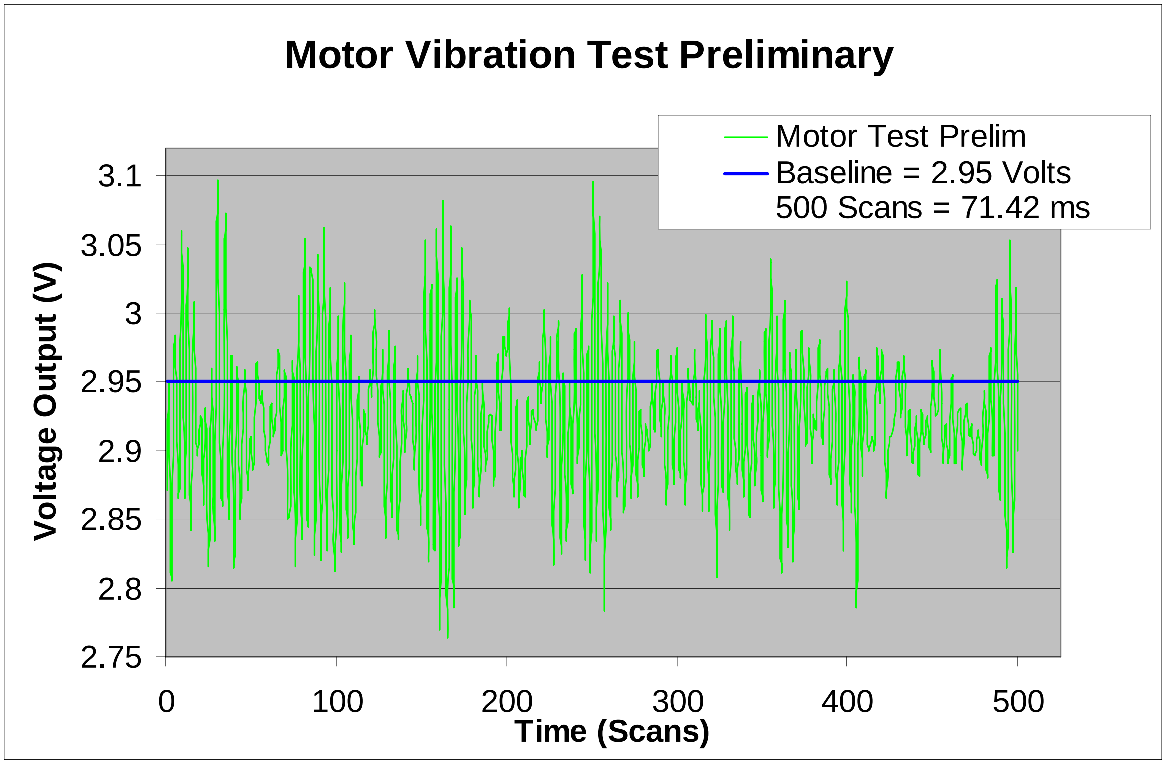
Rod Impact Characterization

Rod impact testing was accomplished in the same way as the base impact testing, yet this time the impact point was the ABS extension rod. Impacting the rod causes a much more pronounced half-sine wave that is seen in (figure Impact_Rod) above. The preliminary values for this test can be seen in (table Impact_Rod) below.

Rod Impact Characterization Results		
Critical Damping Coefficient	Impulse Magnitude	Settling Time
205.2 kg/s	5.073 Volts	186 ms

(Table Impact_Rod)

DC Motor Test



(Figure Vibration_Motor)

Motor Test Characterization

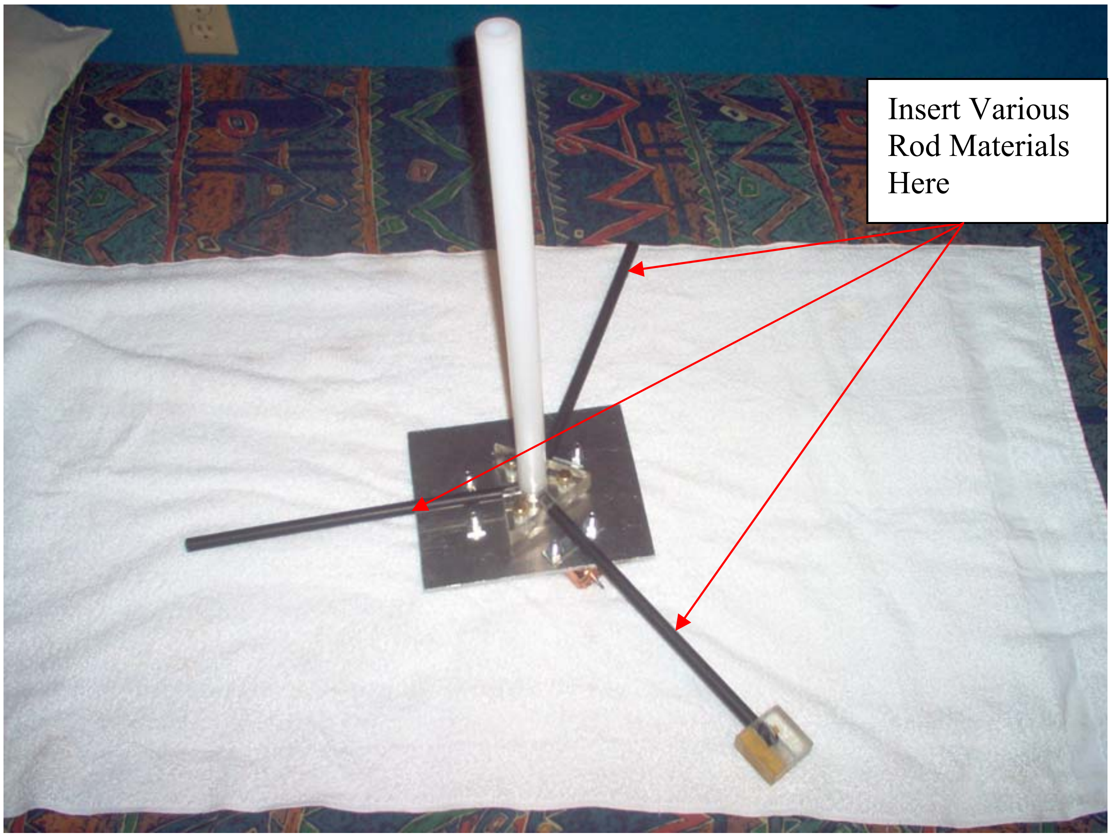
The DC motor test was accomplished in a slightly different manner than the aforementioned testing on the microphone sensor. This test was carried out utilizing a 6 volt DC motor to mimic the actual DC motors that would be found on the RDS robot. The result of this test is that of a resonant frequency wave that can be viewed above in (figure Vibration_Motor). This figure depicts the high frequency excitation of the microphone sensor as well as the relative magnitude of the impulse imposed on the sensor. The characteristic values for the motor vibration test can be seen below in (table Vibration_Motor).

DC Motor Characterization Results	
Microphone Excitation	Impulse Magnitude
1802.5 Hz	3.096 Volts

(Table Vibration_Motor)

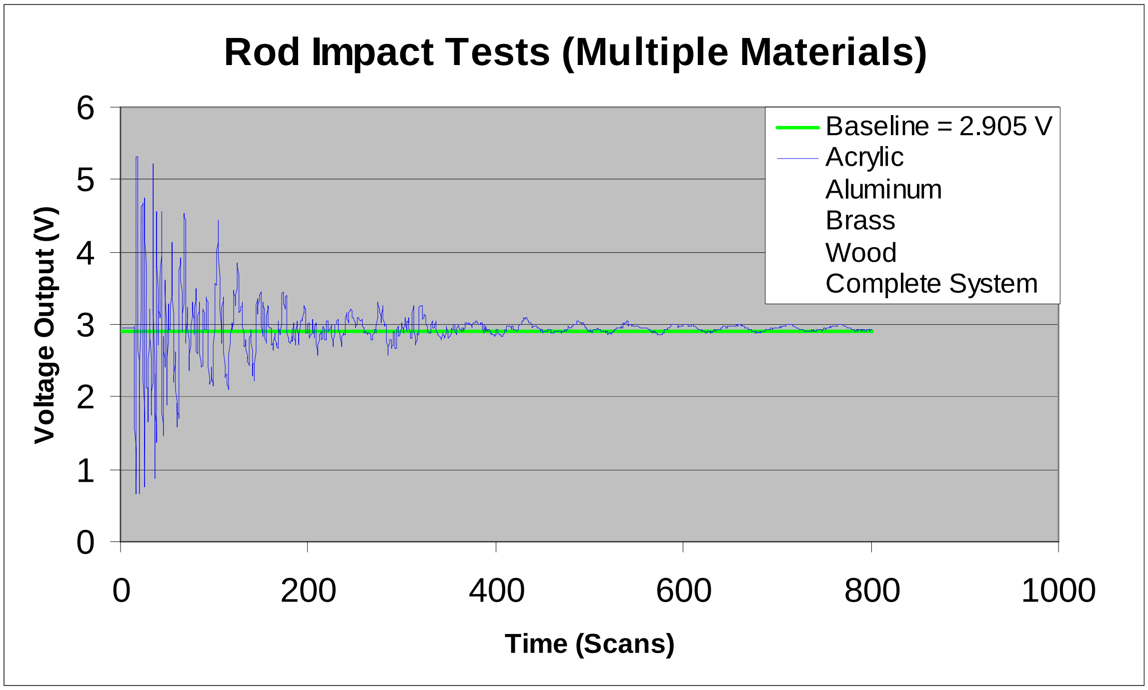
Rod Material Characterization

All preliminary testing was accomplished via the test frame using ABS extension rods as used in the final array designs. A test had to be run to characterize whether or not the ABS rod material was actually the best material to use. To do this, the rod impact test was reused but the test frame was modified to accept different material extension rods as seen in (figure 2-PTest) below. The various other rod materials were acrylic, aluminum, brass, and pine wood, because these are the most common rod materials.

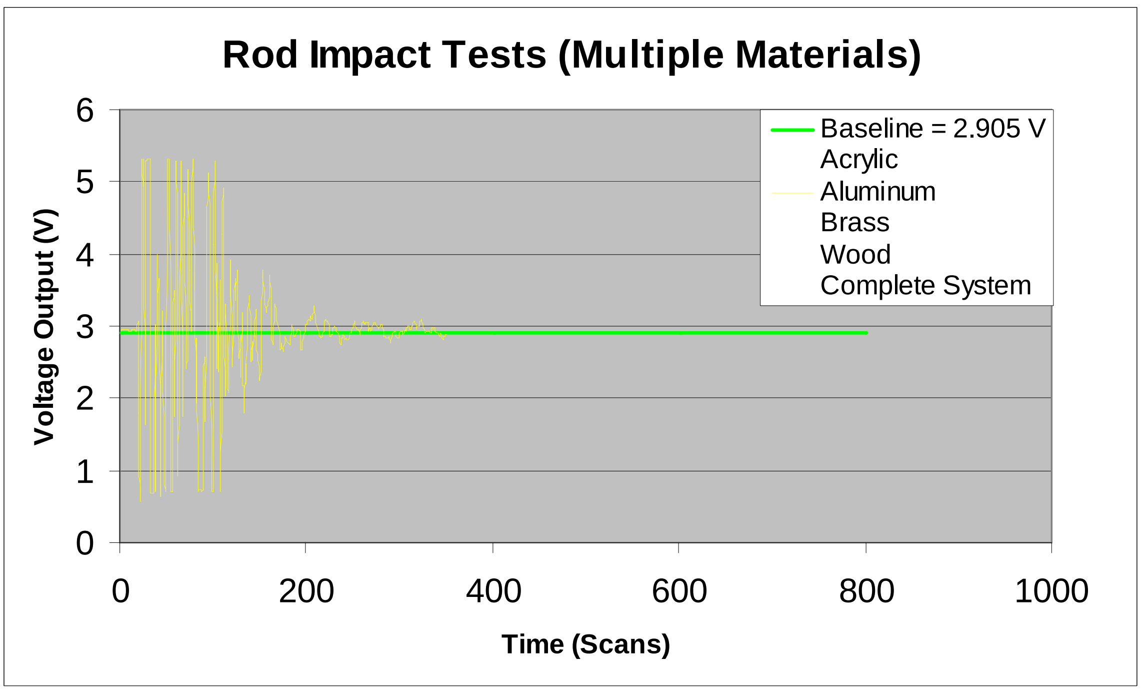


(Figure 2-PTest)

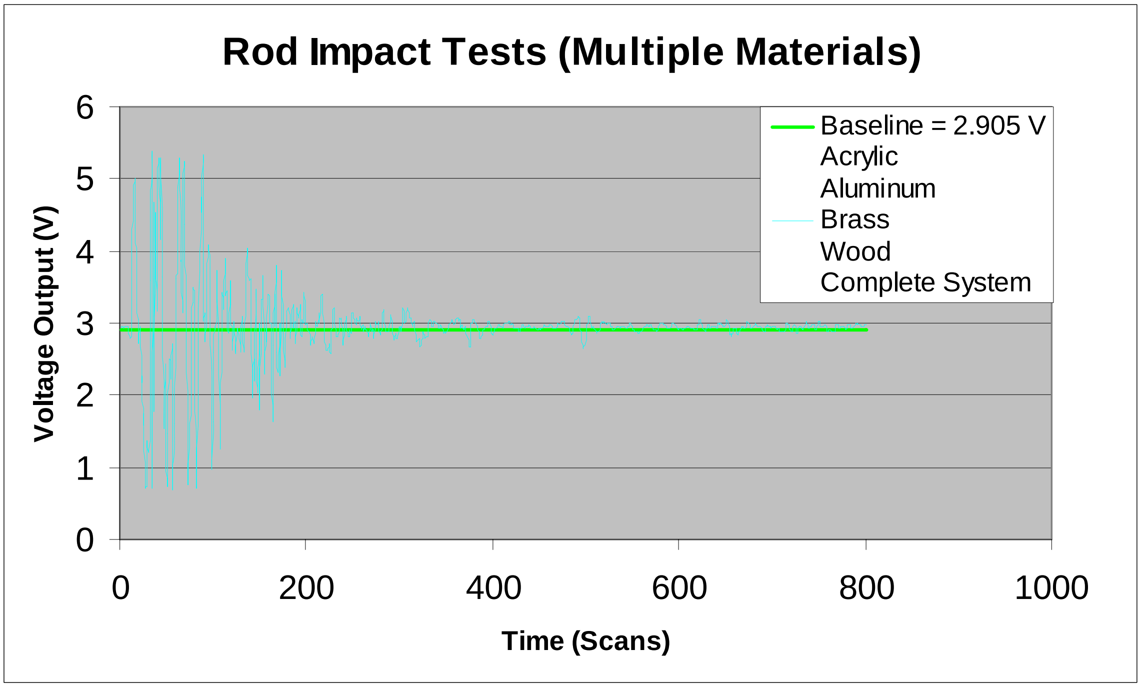
Material Rod Impact Test



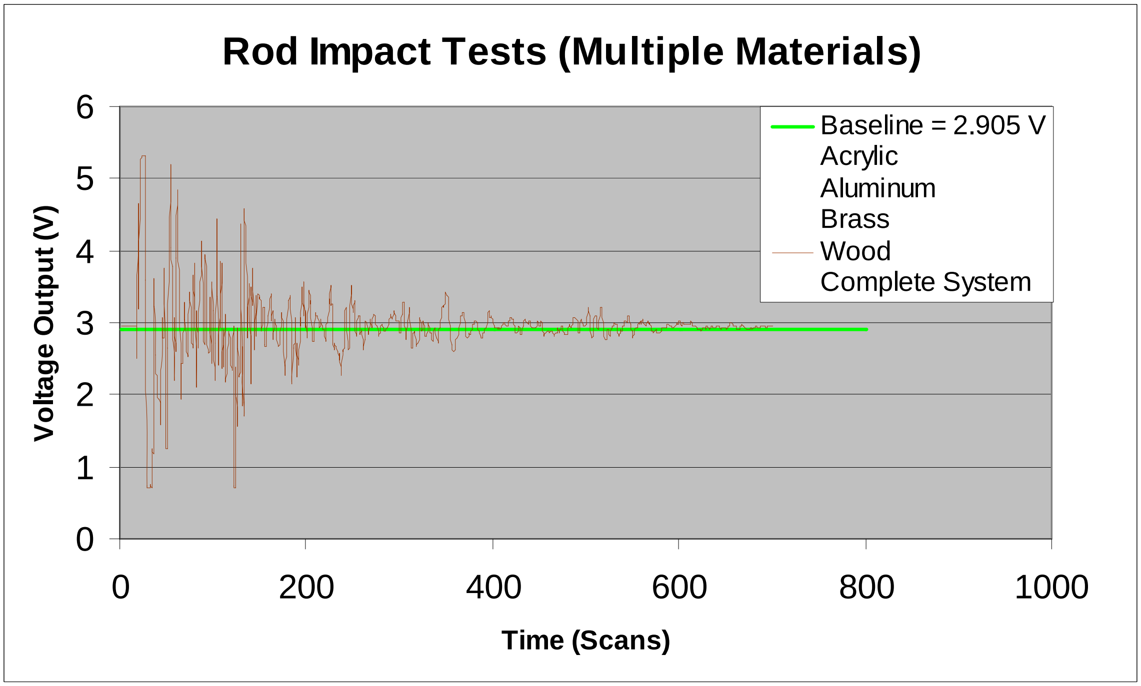
(Figure Mat-Acrylic)



(Figure Mat-Aluminum)



(Figure Mat-Brass)

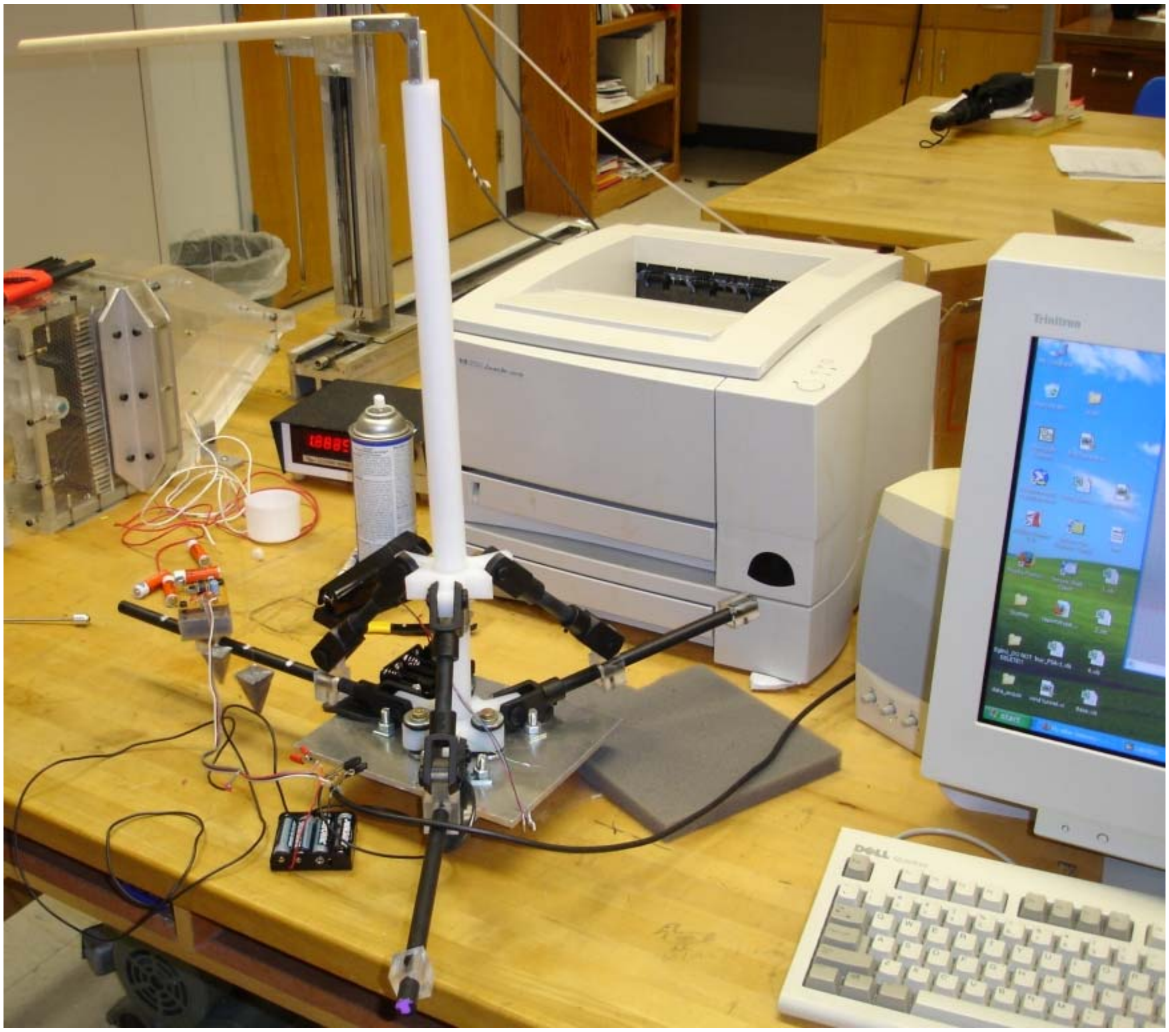


(Figure Mat-Wood)

The graphs above depict the impact results from the various rod materials and the data collected will be used to characterize the ABS rod material against the various materials tested.

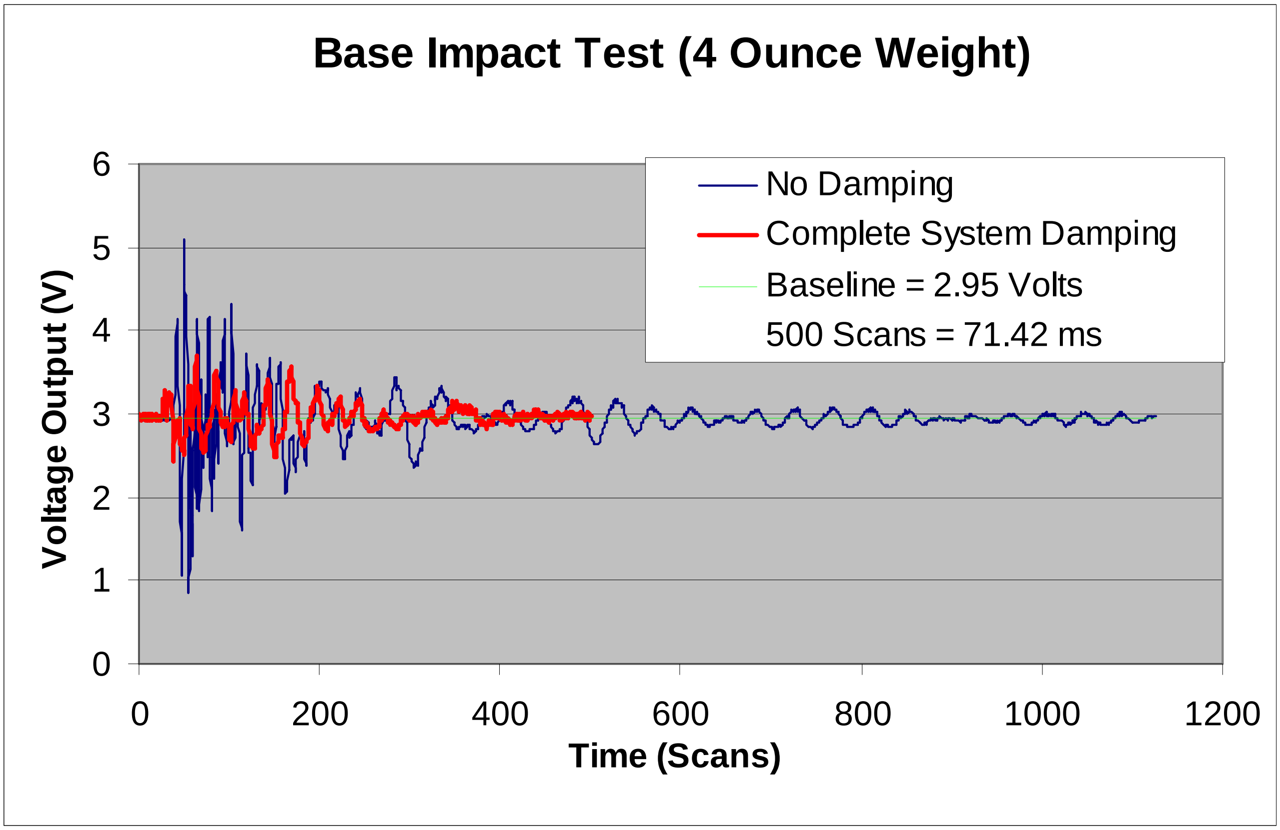
Final Testing

The final testing was accomplished using the completed tetrahedral array with its incorporated vibration isolation and absorption techniques. The final array can be viewed below in (figure 1-Final) already attached to the mock RDS base for the final testing in LabVIEW.



(Figure 1-Final)

Base Impact Test Results



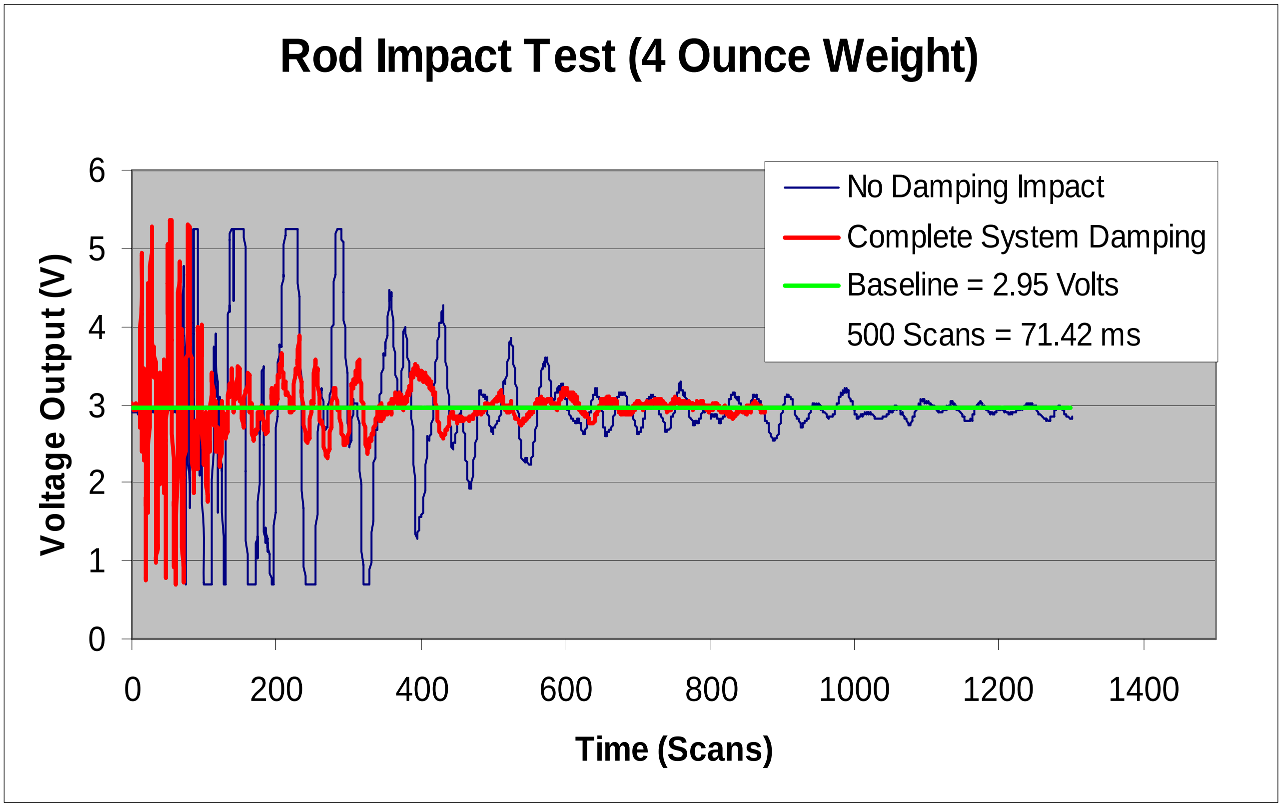
(Figure Impact_Base_2)

Base Impact Results			
Test Type	Critical Damping Coefficient	Impulse Magnitude	Settling Time
Preliminary	866.5 kg/s	5.073 Volts	161 ms
Final	1978.8 kg/s	3.701 Volts	72.4 ms
% Reduction	2 Times Better	65%	55%

(Table Impact_Base_2)

The final results of the base impact testing can be viewed in (figure Impact_Base_2). Comparative analysis between the preliminary and final testing results is outlined in (table Impact_Base_2) above and depicts the advantages that the damping system has over the non-damped system. The critical damping coefficient has been increased 2 fold for the damped system over the non-damped system. This relates to a quicker system response for the damping case and substantial mitigation of vibration that would otherwise be inherent in the system. These findings are evident in the reduction in both the impulse magnitude and settling time of the impulse shock imposed. The damped case shows a substantial 65% reduction in the impulse magnitude and a 55% reduction in the settling time over the non-damped system. The results of the base impact test prove that the techniques used for damping vibration propagation in the array result in much better responses to potential vibration or shock as the platform navigates throughout its environment.

Rod Impact Test Results



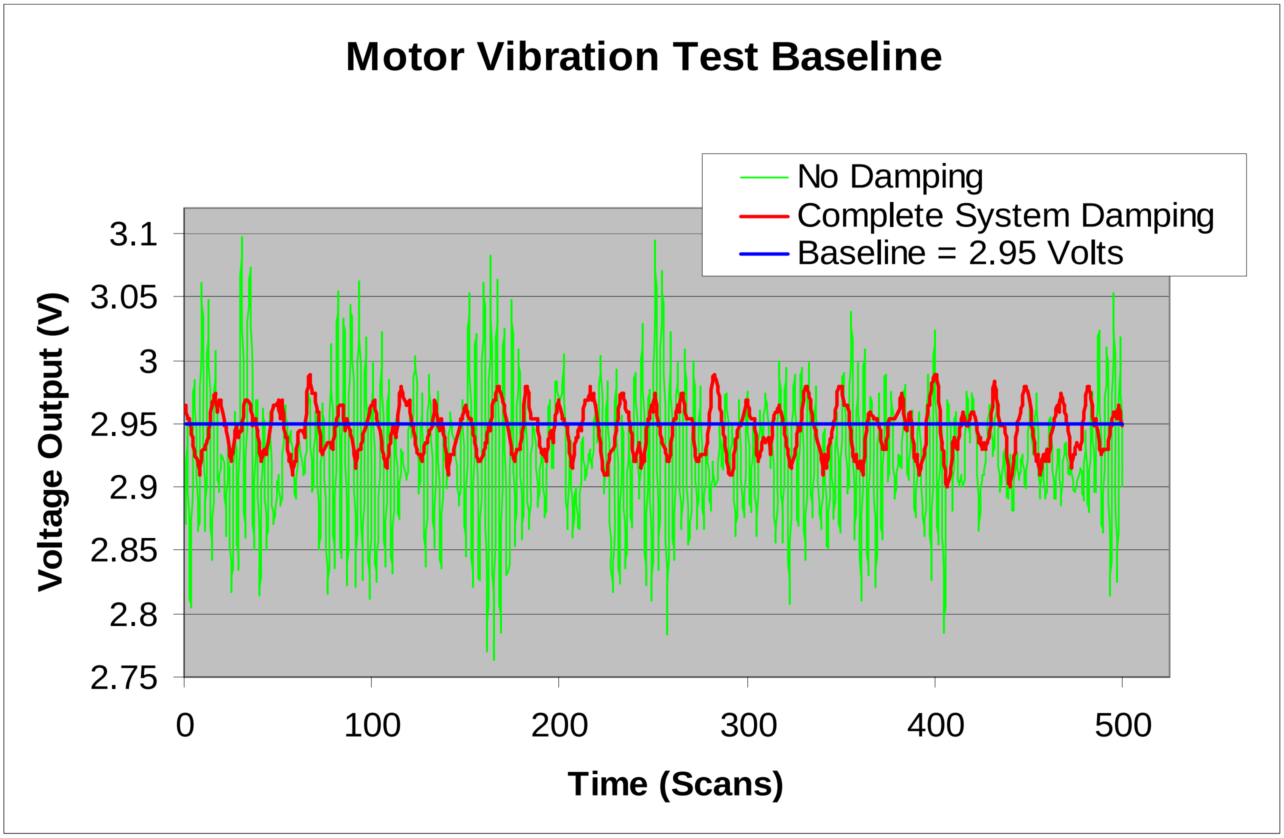
(Figure Impact_Rod_2)

Rod Impact Results			
Test Type	Critical Damping Coefficient	Impulse Magnitude	Settling Time
Preliminary	205.2 kg/s	5.073 Volts	186 ms
Final	1039.3 kg/s	5.361 Volts	125 ms
% Reduction	5 Times Better	4%	33%

(Table Impact_Rod_2)

The main parameter under concern in the direct rod impact testing is the rate at which the system is able to damp out a direct shock impulse to an extension rod on the array platform and this is the critical damping coefficient for the system. Final testing has revealed that the complete system damping case has increased the critical damping coefficient by 5 times over the original, non-damped, system response. This translates into the fact that if and when an extension rod impacts an obstruction it will damp out the impulse introduced at a much quicker rate leaving the microphone sensors undisturbed at a lesser rate. Other improvements to the rod impact test show that the settling time has been reduced by 33% and is directly related to the better damping characteristics of the final array. Due to the fact that the rods are being directly impacted, the impulse magnitude was not expected to have a high reduction and this is evident by only a 4% reduction over the original system. The only way to decrease the impulse magnitude is to further decouple the microphone sensors from the extension rod assemblies.

DC Motor Test Results



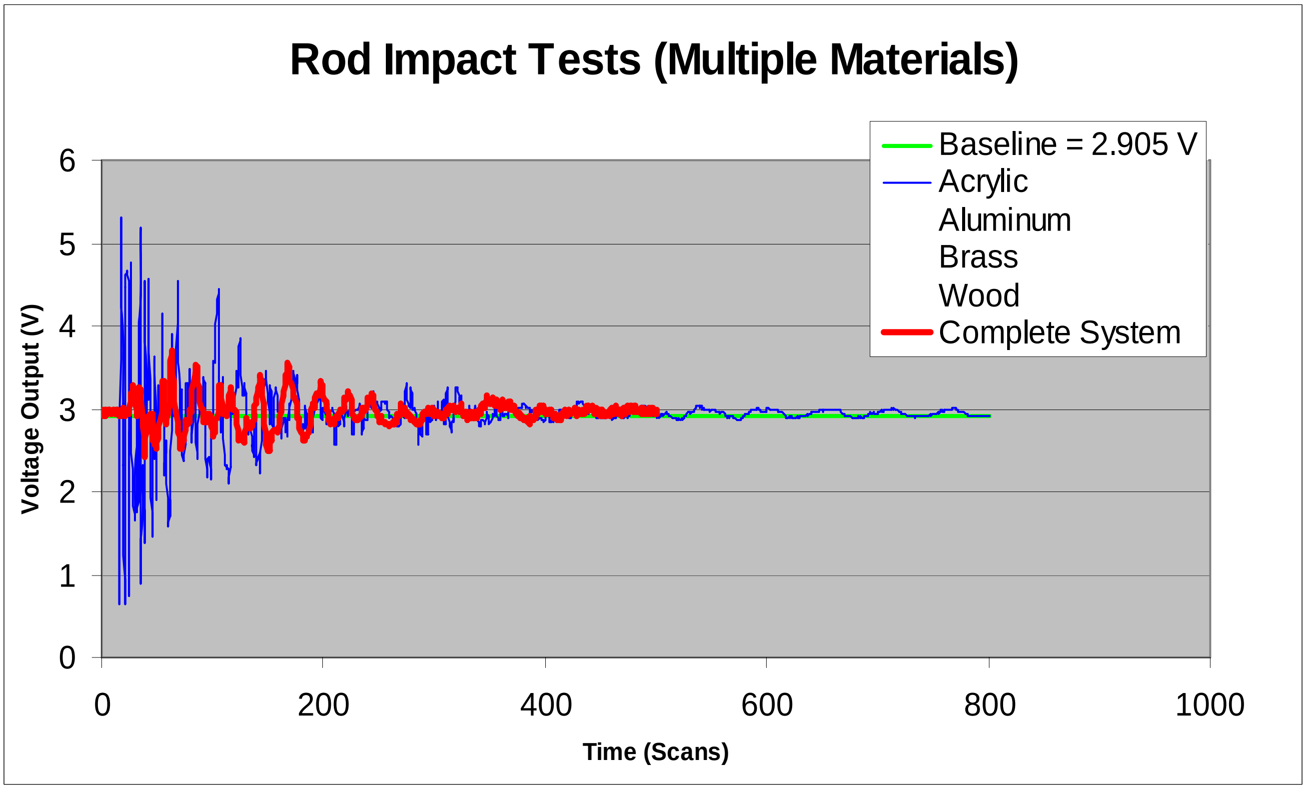
(Figure Vibration_Motor_2)

Design Parameter	No Damping	System Damping	% Reduction
Microphone Excitation	1802.5 Hz	422.8 Hz	76.5 %
Resonance Impulse Magnitude	3.096 Volts	2.988 Volts	74 %

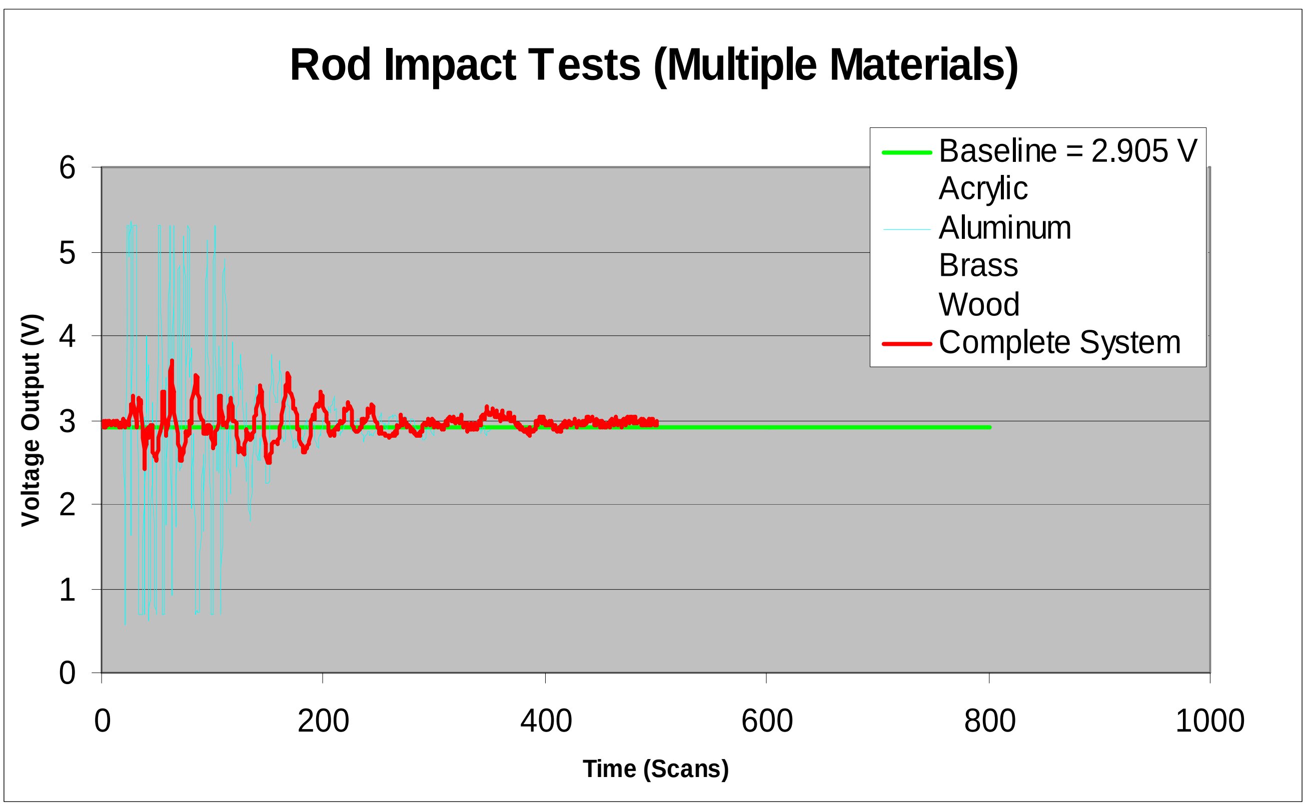
(Table 1-Motor Vibration)

Final testing for the DC motor vibration characterization has resulted in the (figure Vibration_Motor_2) above for the damped system case. The red wave in the above graph depicts the final system damping case and shows a significant decrease in the excitation of the microphone sensor as well as a decrease in the impulse magnitude of the wave. Comparisons between preliminary and final DC motor testing has revealed that the completed system was able to damp out 74% of the impulse magnitude being received by the microphone sensor. Excitation of the microphone sensor has also been reduced by 76.5%. The reduction in the impulse magnitude received by the microphone sensor in the final system is evident that the damping measures utilized provide a very adequate means of vibration mitigation to the system as a whole.

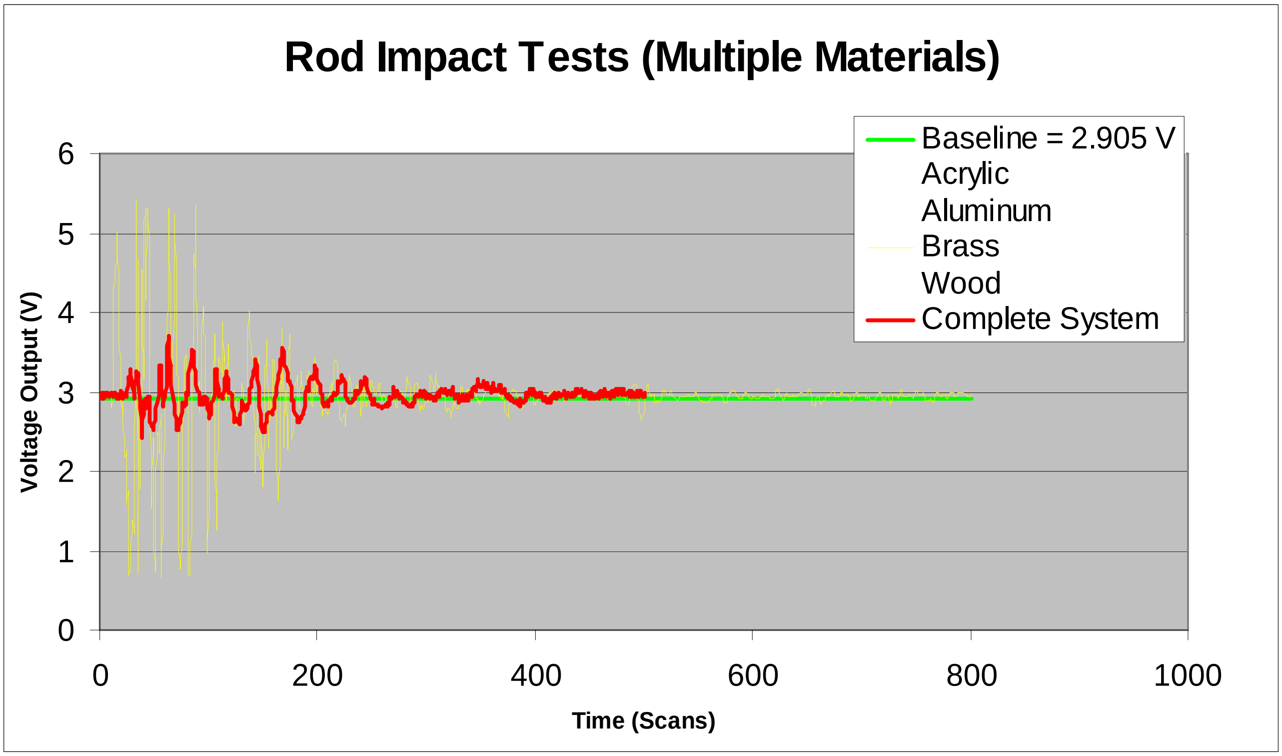
Material Rod Impact Test Results



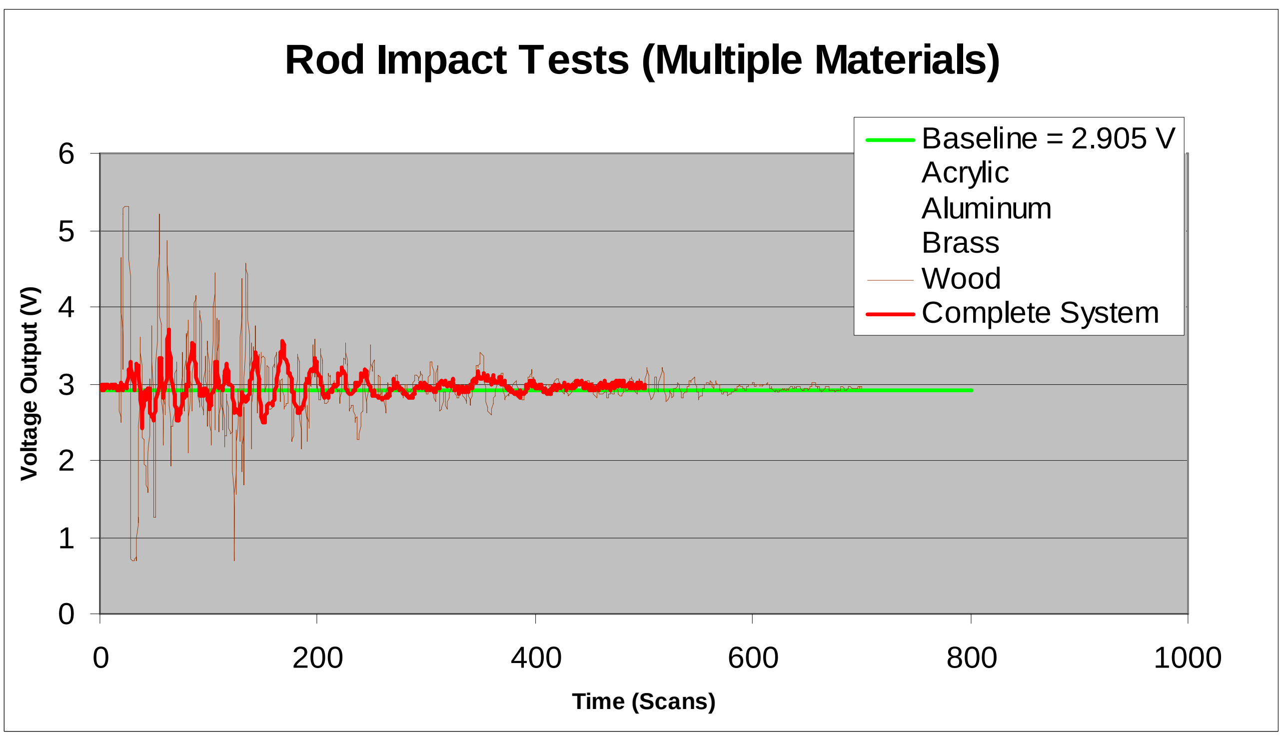
(Figure Mat-Acrylic)



(Figure Mat-Aluminum)



(Figure Mat-Brass)



(Figure Mat-Wood)

Final testing was compared to the preliminary rod impact tests on the different rod material types to get a comparison between the most common types of rod materials. This testing comparison resulted in the above figures for the different rod materials versus using ABS as the rod material. The comparative analysis can be viewed in the below (table 1-Rod_Comparison).

Material	Acrylic	Aluminum	Brass	Wood (Pine)	ABS System
Settling Time	114 ms	50.7 ms	110 ms	100 ms	72.4 ms
Impulse Magnitude	5.308 V	5.308 V	5.308 V	5.303 V	3.701 V

(Table 1-Rod_Comparison)

The final comparison between the different rod materials and the ABS rod material used has shown that ABS is the more practical rod material for the application of vibration damping and mitigation. The ABS has a substantially less shock impulse magnitude than any other rod material considered and thus is a better damper. The settling time for the ABS is also less in all cases except for that of Aluminum. The reason for this is that the Aluminum has a very high sound conduction factor and thus the impulse will resonate more quickly through the Aluminum rod than the ABS. The only issue that this presents is that the Aluminum is not damping the impulse while the ABS rod is and this is where the greater settling time for the ABS occurs. Overall, the ABS rod is the best rod material for this application in terms of its damping characteristics as well as its lightweight and low cost benefits.

Future Recommendations

The results of the final design on vibration mitigation have substantial benefits for the acoustic sensor eye integration into the RDS platform. These results are however by no means the only way by which to accomplish the required constraints set forth by the project objectives. This relates to further work that may be done to achieve even better or simpler methods of vibration damping characteristics. Some recommendations are outlined here as merely ideas that may or may not be followed up on to increase the results found through this document.

1.) Adapter Plate Modification

   i. This plate is merely an adapter between the tetrahedral array frame and RDS platform. No specifications were given in terms of dimensions for the RDS mounting location at the time of this projects' completion. Further specifications may allow for a better adapter plate to be constructed in the future.

2.) RDS to Tetrahedral Array Damping

   i. No physical substrate between the actual mounting points to the RDS platform was introduced into the final design due to not having the required specifications for this operation. A substrate may be added at a future time to the mounting location to further enhance the damping characteristics of the array frame.

3.) Redesign of Microphone Sensor Mounts

   i. The original microphone mount was designed and built around a supplied prototype microphone. A later production type microphone was introduced after manufacturing of the original mounts and an adapter plate was design and implemented. A redesign of the microphone mounting plate will greatly reduce the complexity of this portion of the array.

4.) Damping Material Research

   i. The materials used in the final array frames are by no means the only damping type materials that may be used. Further research into other materials may reveal a material that is better suited for vibration mitigation.

5.) Power Conservation

   i. Research into a different type of actuation system to reduce the power consumption for deploying and retracting the acoustic eye array.
   ii. Also research into reducing the friction coefficient of moving parts of the array to reduce force required for motion.

Conclusions

The senior design team has made great strides in implementing all constraints set forth by Eglin AFRL/MN and has come up with designs that suit their needs. The team was able to identify how to damp out mechanical vibration in a rigid frame and apply that knowledge to build a suitable frame housing for both the RDS platform and VEX™ robot. Once all the constraints were addressed a design matrix was designed to choose the best idea that was generated during the ideation phase. After the design was chosen, the design work began using the research obtained that was pertinent to the constraints. This research included finding various materials suitable for vibration suppression as well as other components that adhered to the design constraints of a lightweight structure of low cost. Materials such as UHMW-PE (Ultra High Molecular Weight Polyethylene) and ABS (Acrylanitrile Butadiene Styrene) were chosen for the structure of the frame. These materials had desirable vibration mitigation characteristics coupled with structural rigidity at a low weight and low cost. Sorbothane® and acoustic foam were chosen for the vibration isolation and absorption aspect of the frame structure. Igus® Clevis joints were chosen for the collapsibility aspect of the tetrahedral frame structure and a Z2684X-V stepper motor was chosen for the actuation of the final tetrahedral frame. From this point, testing procedures were developed to test for the properties of the various components chosen to obtain their vibration damping abilities as a system.

A test frame was built from the selected materials to serve as a test bed for the design. Three tests were run on the test and final frames, a DC motor test, base impact test, and rod impact test. The results of the preliminary testing were used as a baseline to characterize the vibration inherent in the original acoustic eye sensor system for later comparison. The final frames were then built and tested to prove that a mitigation of the inherent system vibration had in fact occurred. The results of the final testing proved that a reduction in the impulse magnitude had occurred. The impulse magnitude for the base impact test was reduced by as much as 65% over the original characterized system vibration. The DC motor test showed 74% reduction in the impulse magnitude response of the microphone sensor of the damped system. This translates into the fact that the vibration damping techniques were reducing the magnitude of the motor excitation of the sensor which results in less interference for the sensor during navigation operations. The rod impact tests proved that ABS rod material was the clear choice when considering other common rod materials as outlined in the **(Testing)** section of this document. The critical damping coefficient was also increased by as much as 5 times over the original damping rate for the non-damped system. This increase in the damping rate shows that direct rod impacts will disturb the microphone sensors at a substantially less rate than that of the original, non-damped system. The overall findings from all the testing proves that vibration techniques can be incorporated into a lightweight and cheap design. With this in mind, a substantial decrease in the acoustic eye sensors overall error propagation is lessened by the use of vibration damping techniques.

Overall, the final designs for both the tetrahedral array frame and the T-base array adhere to all constraints brought forth by Eglin AFRL/MN. The two frames consist of a tetrahedral geometry with the microphone sensors facing upward in the horizontal plane. They both incorporate the same vibration damping characteristics in terms of both isolation and absorption. The microphone mounting distances were met for both array types and both arrays include fine tune adjustment methods for the microphone-to-microphone distances. Only off-the-shelf components were utilized in the designs for both arrays and this helped with adhering to a low cost design. Both arrays are lightweight with respect to the platform in which they are to be mounted. The tetrahedral array weighs approximately 2 pounds, while the T-base array weighs merely ounces. All design parameters were met and complete analysis proves that the arrays do indeed adhere to the necessities of the projected user.

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Appendix A: Calculations

Pendulum Momentum Calculation

Given:

$m_1 := 0.125\text{lb}$ $m_2 := 0.25\text{lb}$ $h_1 := 20\text{in}$ $h_2 := 5.86\text{in}$

Find:

Momentum of mass just before impact with rod.

Solution:

Conservation of Energy: No work

Solving the energy equations for velocity at the bottom of the pendulum's swing yields:

$v_1 := \sqrt{2 \cdot g \cdot h_1}$ $v_2 := \sqrt{2 \cdot g \cdot h_2}$ $v_1 = 10.356 \frac{ft}{s}$ $v_2 = 5.606 \frac{ft}{s}$

Momentum:

There are two masses being used (m1 and m2), and two heights (h1 and h2) which result in two corresponding velocities (v1 and v2 above). This combination yields four possible values of momentum shown below:

$M_{11} := m_1 \cdot v_1$ $M_{11} = 1.295\text{lb} \cdot \frac{ft}{s}$ **Momentum of the lower weight dropped at 90 deg.**

$M_{12} := m_1 \cdot v_2$ $M_{12} = 0.701\text{lb} \cdot \frac{ft}{s}$ **Momentum of the lower weight dropped at 45 deg.**

$M_{21} := m_2 \cdot v_1$ $M_{21} = 2.589\text{lb} \cdot \frac{ft}{s}$ **Momentum of the heavier weight dropped at 90 deg.**

$M_{22} := m_2 \cdot v_2$ $M_{22} = 1.401\text{lb} \cdot \frac{ft}{s}$ **Momentum of the heavier weight dropped at 45 deg.**

Actuator Force Calculations

$V_{slider} := 2.741\ in^3$

$V_{collar} := 0.7762\ in^3$

$V_{con} := 0.547\ in^3$ **Volumes of Various Components**

$V_{rod} := 1.692\ in^3$

$V_{clevis} := 1.0112\ in^3$

$\rho_{ABS} := 1.052 \cdot 10^{-3} \frac{kg}{cm^3}$

$\rho_{Plexi} := 1.162 \cdot 10^{-3} \frac{kg}{cm^3}$ **Density of Various Components**

$\rho_{igumid} := 1.4 \cdot 10^{-3} \frac{kg}{cm^3}$

$\mu := 0.1$ **Friction coefficient for UHMW-PE**

$m_{slider} := \rho_{Plexi} \cdot V_{slider}$

$m_{slider} = 0.052\ kg$ **Mass of Slider**

$m_{collar} := \rho_{Plexi} \cdot V_{collar}$

$m_{collar} = 0.015\ kg$ **Mass of 90 degree collar link**

$m_{con} := \rho_{ABS} \cdot V_{con}$

$m_{con} = 9.43 \times 10^{-3}\ kg$ **Mass of Connecting Arm**

$m_{rod} := \rho_{ABS} \cdot V_{rod}$

$m_{rod} = 0.029\ kg$ **Mass of Extending Rod**

$m_{clevis} := \rho_{igumid} \cdot V_{clevis}$

$m_{clevis} = 0.023\ kg$ **Mass of Clevis Assembly**

$m_{mic} := 0.06804\ kg$

$m_{mic} = 0.068\ kg$ **Assumed mass of one speaker assembly**

Now to calculate the force of one Rod assembly:

$F_{mic} := m_{mic} \cdot g$

$F_{mic} = 0.667N$

$F_{rod} := m_{rod} \cdot g$

$F_{rod} = 0.286N$

$F_{con} := m_{con} \cdot g$

$F_{con} = 0.092N$

$F_{collar} := m_{collar} \cdot g$

$F_{collar} = 0.145N$

$F_{clevis} := m_{clevis} \cdot g$

$F_{clevis} = 0.228N$

$$T_{rod} := \frac{F_{clevis} \cdot 3 + F_{collar} + F_{con} + F_{rod} + F_{mic}}{\cos(47deg)}$$

$T_{rod} = 2.747N$ **Force Required to Lift one Rod Assembly.**

$T_{rod\_assy} := T_{rod} \cdot 3$

$T_{rod\_assy} = 8.24N$ **Force Required to lift entire rod assemblies**

$F_{required} := T_{rod\_assy} + F_{collar}$

$F_{required} = 8.385N$ **Force Required to fold assembly from fully extended position without factoring in friction.**

$F_{Friction} := \mu \cdot F_{required}$

$F_{Friction} = 0.838N$ **Force of friction that must be overcome throughout entire frame**

$F_{actuator} := F_{required} + F_{Friction}$

$F_{actuator} = 9.223N$ **Force Required to fold structure, therefore a 10 Newton actuator will be sufficient to deploy and retract frame.**

Material Usage and Percent Savings Calculations

Four-Bar Slider Housing Rod Material

$Extending := 11.55 \cdot in$

$Center := 16.33 \cdot in$

$Total := 3 \cdot Extending + Center$

$Total = 50.98\,in$

Other Design Alternatives

$Extending\_O := 20 \cdot in$

$Center\_O := 17.32 \cdot in$

$Total\_O := 3 \cdot Extending\_O + Center\_O$

$Total\_O = 77.32\,in$

Percent Savings in Material

$$\%\_Savings := 1 - \frac{Total}{Total\_O}$$

$\%\_Savings = 34.07\,\%$

Estimated Price Calculation for each design type production

$dollars := 1$

$Each := 1$

$Composite\_Shaft := 4.56 \cdot \frac{dollars}{12in}$ **Price per unit of composite shaft**

$Shaft\_Screw := 20.76 \cdot \frac{dollars}{Each}$ **Price of Shaft Screw**

$Slider := 5.60 \cdot dollars$ **Price to produce slider mechanism**

$Clevis := 8.36 \cdot \frac{dollars}{Each}$ **Price of Clevis Joint Assembly**

$Pivot := 17.39 \cdot \frac{dollars}{Each}$ **Price of Pivot Link**

$Pulleys := 7.59 \cdot \frac{dollars}{Each}$ **Price of Pulleys**

$Cables := 27.75 \cdot \frac{dollars}{Each}$ **Price per cable**

Tetrahedron over Tetrahedron

$C := 1 \cdot Each \cdot Shaft\_Screw$ **Shaft screw total cost**

$E := (6 \cdot 20in) \cdot Composite\_Shaft$ **Composite extending shaft total cost**

$Clevis\_N := 6 \cdot Each \cdot Clevis$ **Clevis Joints Required total cost**

$P := 3 \cdot Each \cdot Pivot$ **Pivot Joints total cost**

$Total\_Cost := C + E + Clevis\_N + P + Slider$

$Total\_Cost = 174.29 dollars$ **Total System Cost**

Four-Bar Slider Folding Down

$C\_S := 16.33in \cdot Composite\_Shaft$ **Composite center shaft total cost**

$E\_R := (3 \cdot 11.55in) \cdot Composite\_Shaft$ **Composite extending shaft total cost**

$C\_R := (3 \cdot 6in) \cdot Composite\_Shaft$ **Composite connecting shaft total cost**

$Cl\_N := 6 \cdot Each \cdot Clevis$ **Clevis Joints Required total cost**

$Pt\_N := 3 \cdot Each \cdot Pivot$ **Pivot Joints total cost**

$Total\_Cost := C\_S + E\_R + C\_R + Pt\_N + Slider + Cl\_N$

$Total\_Cost = 134.14 dollars$ **Total System Cost**

Jointed Tetrahedral Slider Opening Up

$Cr\_S := 33in \cdot Composite\_Shaft$	**Composite center shaft total cost**
$Ex\_R := (3 \cdot 20in) \cdot Composite\_Shaft$	**Composite extending shaft total cost**
$Co\_R := (3 \cdot 6in) \cdot Composite\_Shaft$	**Composite connecting shaft total cost**
$Cle\_N := 6 \cdot Each \cdot Clevis$	**Clevis Joints Required total cost**
$Pi\_N := 3 \cdot Each \cdot Pivot$	**Pivot Joints total cost**

$Total\_Cost := Cr\_S + Ex\_R + Co\_R + Pi\_N + Slider + Cle\_N$

$Total\_Cost = 150.11 dollars$ **Total System Cost**

Cable-Controlled Tetrahedral Deployment

$C\_Sh := 33in \cdot Composite\_Shaft$	**Composite center shaft total cost**
$Ex\_Ro := (3 \cdot 20in) \cdot Composite\_Shaft$	**Composite extending shaft total cost**
$P := 9 \cdot Each \cdot Pulleys$	**Pulleys required for system cost**
$C := 3 \cdot Each \cdot Clevis$	**Clevis Joints Required total cost**
$Cable := 3 \cdot Each \cdot Cables$	**Cables Required Total Cost**

$Total\_Cost := C\_Sh + Ex\_Ro + C + P + Cable$

$Total\_Cost = 211.98 dollars$ **Total System Cost**

Speed of Sound conduction through a material calculations

The speed of sound in a solid

$$\nu = \left(\frac{E}{\rho}\right)^{\frac{1}{2}}$$

Where E is Young's modulus and rho is the density of the matrial.

The following values were taken from global polymers inc. for their materials and matweb. Several sources were found for these modulus and density numbers and these values were consistent with all the other sources.

Young's Modulus (E) for The Selected Materials:

$E_{polycarb} := 320000\,psi$ **Polycarbonate**

$E_{abs} := 270000\,psi$ **ABS (Acrylanitrile butadiene Styrene)**

$E_{carbon} := 696192\,psi$ **Carbon**

$E_{graphite} := 2103080\,psi$ **Graphite**

$E_{uhmw} := 90000\,psi$ **UHMW-PE (Ultra High Molecular Weight - Polyethelene)**

$E_{peek} := 450000\,psi$ **PEEK (Polyetheretherketone)**

Density for The Selected Materials:

$\rho_{polycarb} := 0.0440\frac{lb}{in^3}$ **Polycarbonate**

$\rho_{abs} := .04\frac{lb}{in^3}$ **ABS (Acrylanitrile butadiene Styrene)**

$\rho_{carbon} := .0813\frac{lb}{in^3}$ **Carbon**

$\rho_{graphite} := 0.0643\frac{lb}{in^3}$ **Graphite**

$\rho_{uhmw} := 0.0336\frac{lb}{in^3}$ **UHMW-PE (Ultra High Molecular Weight - Polyethelene)**

$\rho_{peek} := 0.047\frac{lb}{in^3}$ **PEEK (Polyetheretherketone)**

The Speed of Sound conduction in The Selected Material:

$$v_{polycarb} := \left(\frac{E_{polycarb}}{\rho_{polycarb}}\right)^{\frac{1}{2}}$$

$$\boxed{v_{polycarb} = 4415.82\frac{ft}{s}}$$ **Polycarbonate**

$$v_{abs} := \left(\frac{E_{abs}}{\rho_{abs}}\right)^{\frac{1}{2}}$$

$$\boxed{v_{abs} = 4254.163\frac{ft}{s}}$$ **ABS (Acrylanitrile butadiene Styrene)**

$$v_{carbon} := \left(\frac{E_{carbon}}{\rho_{carbon}}\right)^{\frac{1}{2}}$$

$$\boxed{v_{carbon} = 4791.61\frac{ft}{s}}$$ **Carbon**

$$v_{graphite} := \left(\frac{E_{graphite}}{\rho_{graphite}}\right)^{\frac{1}{2}}$$

$$\boxed{v_{graphite} = 9364.501\frac{ft}{s}}$$ **Graphite**

$$v_{uhmw} := \left(\frac{E_{uhmw}}{\rho_{uhmw}}\right)^{\frac{1}{2}}$$

$$\boxed{v_{uhmw} = 2679.871\frac{ft}{s}}$$ **UHMW-PE (Ultra High Molecular Weight - Polyethelene)**

$$v_{peek} := \left(\frac{E_{peek}}{\rho_{peek}}\right)^{\frac{1}{2}}$$

$$\boxed{v_{peek} = 5066.634\frac{ft}{s}}$$ **PEEK (Polyetheretherketone)**

Final Test Results Calculations

4 Ounce Rod Impact Analysis:

$ms := \frac{1}{1000} s$

$\rho := 0.04 \frac{lb}{in^3}$

$r := 0.5 \cdot in$

$L := 11.55 \cdot in$

$v := \pi \cdot r^2 \cdot L$

$v = 148.653 cm^3$

$m := \rho \cdot v$

$m = 0.363 lb$ **Mass of Rod**

4 Ounce rod Impact with NO Substrate:

$\tau := 10.08 \cdot ms$ **Period**

$\omega_n := \frac{2\pi}{\tau}$

$\omega_n = 623.332 \frac{rad}{s}$ **Natural frequency**

$c_c := 2 \cdot m \cdot \omega_n$

$c_c = 205.186 \frac{kg}{s}$ **Critical Damping Coefficient** **This value provides the quickest rate at which the amplitude of the oscillation will reach zero.**

Frequency Calculation:

$f := \frac{1}{\tau}$

$f = 99.206 Hz$ **Frequency of Non-Substrate System**

4 Ounce rod Impact with Substrate:

$\tau := 1.99 \cdot ms$ **Period**

$\omega_n := \frac{2\pi}{\tau}$

$\omega_n = 3.157 \times 10^3 \frac{rad}{s}$ **Natural frequency**

$c_c := 2 \cdot m \cdot \omega_n$

$c_c = 1039.332 \frac{kg}{s}$ **Critical Damping Coefficient** **This value provides the quickest rate at which the amplitude of the oscillation will reach zero.**

Frequency Calculation:

$f := \frac{1}{\tau}$

$f = 502.513 Hz$ **Frequency of Substrate System**

Percent Increase in Damping Coefficient Rod Impact 4 ounce:

$c_{c\_Base} := 205.186 \frac{kg}{s}$ **Baseline Damping Coefficient**

$c_{c\_Sub} := 1039.332 \frac{kg}{s}$ **Substrate Damping Coefficient**

$$\text{Effectiveness} := \frac{c_{c\_Sub} - c_{c\_Base}}{c_{c\_Sub}}$$

$\text{Effectiveness} = 80.3\%$ **Increase in damping effectiveness**

Motor Frequency Analysis:

$$ms := \frac{1}{1000}s$$

$\tau_1 := 0.5548ms$ **Period of no damping motor test**

$$f_1 := \frac{1}{\tau_1}$$

$f_1 = 1802.451Hz$ **Frequency of no damping motor Test**

$\tau_2 := 2.365ms$

$$f_2 := \frac{1}{\tau_2}$$

$f_2 = 422.833Hz$ **Frequency of damped motor test**

Determination of frequency reduction:

$$f_{red} := \frac{f_1 - f_2}{f_1}$$

$f_{red} = 76.541\%$ **Percent decrease in frequency received by microphone sensor**

Motor Voltage Magnitude Analysis:

$M_1 := 3.096V$ **Voltage output of undamped motor test**

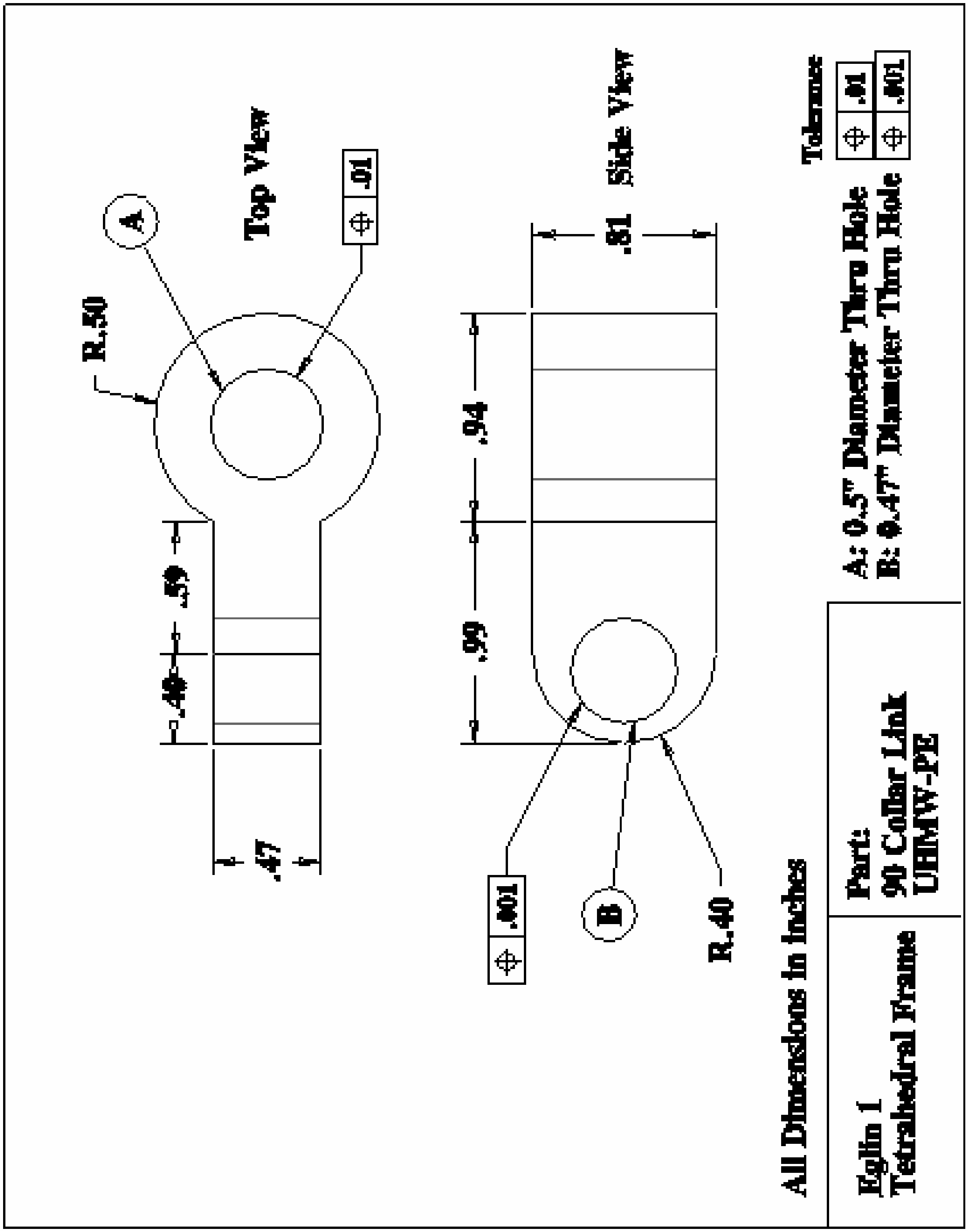
$M_2 := 2.988V$ **Voltage output of damped motor test**

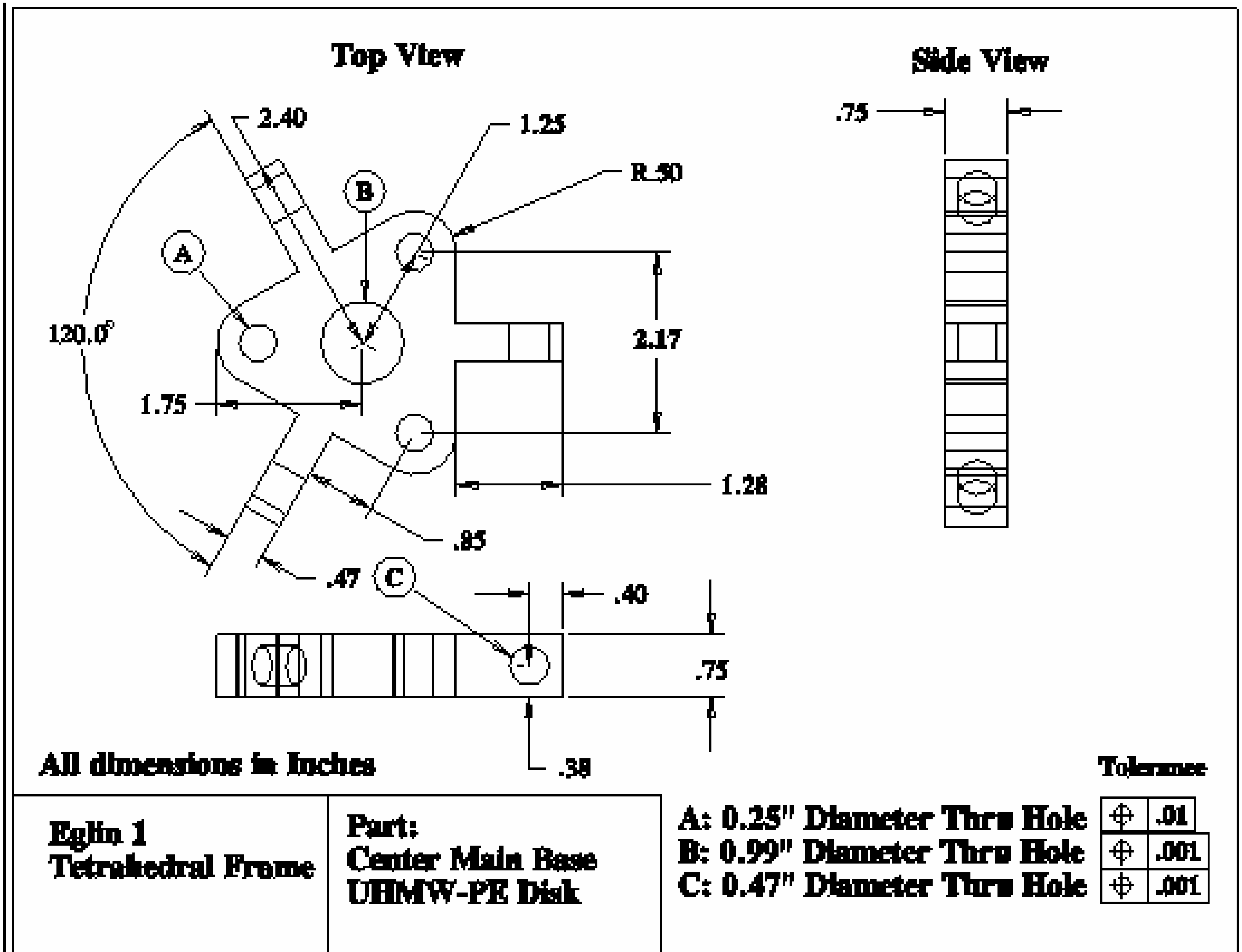
$V_{not} := 2.95V$ **Baseline mic voltage**

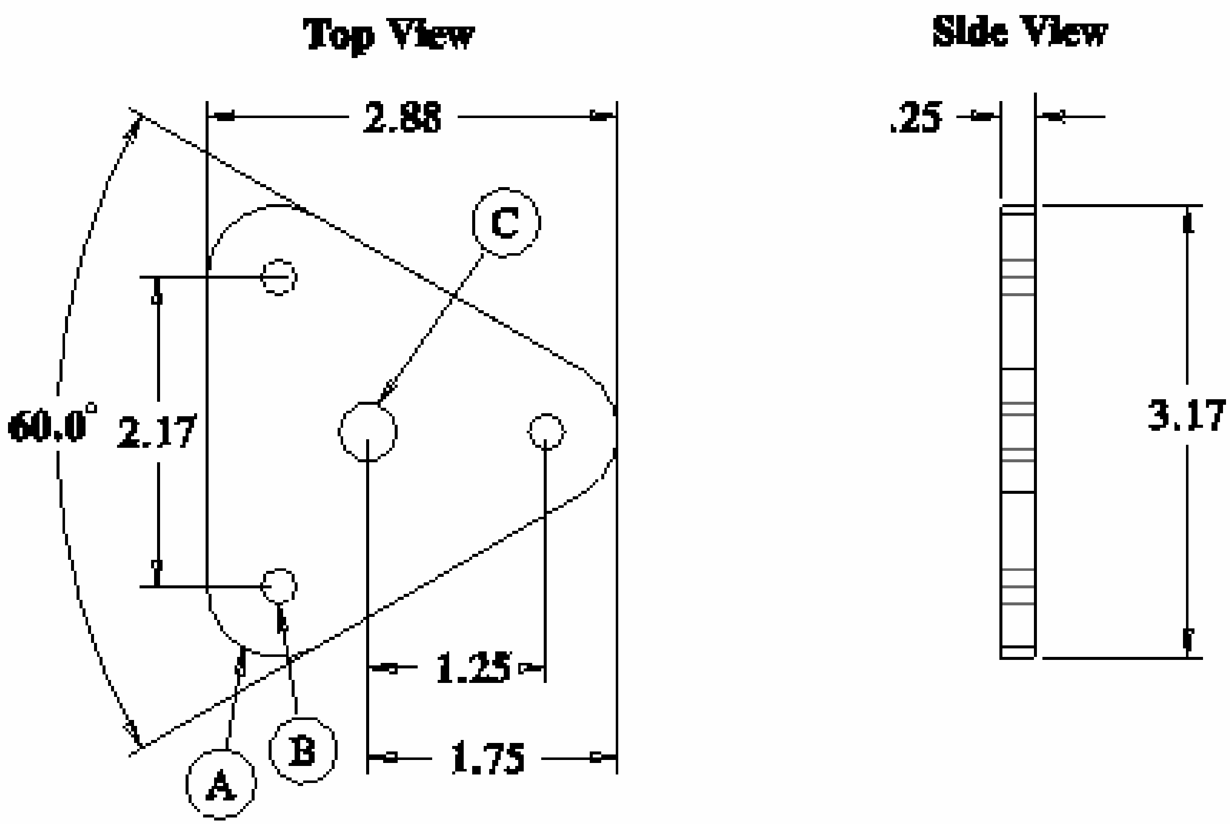
$$Mag_{Red} := \frac{(M_1 - V_{not}) - (M_2 - V_{not})}{(M_1 - V_{not})}$$

$Mag_{Red} = 73.973\%$ **Percent reduction in the magnitude of the voltage output of the microphone sensor**

Appendix B: CAD Drawings


R.50
A
Top View
.01
.47
.40
.59
.94
.99
.81
Side View
.001
B
R.40
Tolerance
.01
.001
A: 0.5" Diameter Thru Hole
B: 0.47" Diameter Thru Hole
All Dimensions In Inches
Eglin 1
Tetrahedral Frame
Part:
90 Collar Link
UHMW-PE

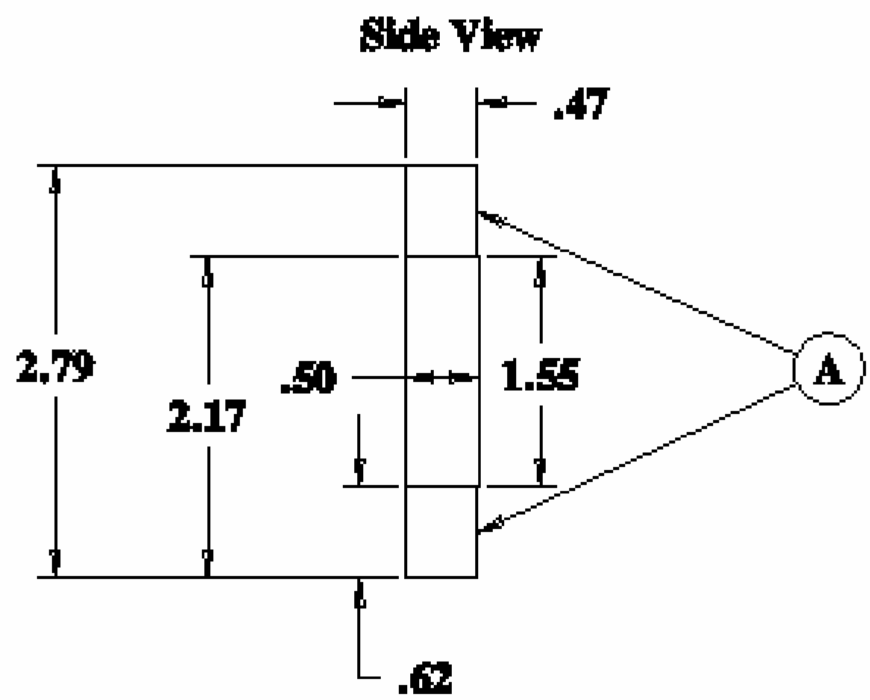

Top View
Side View
2.40
1.25
R.50
B
A
120.0°
2.17
1.75
1.28
.85
.47
C
.40
.75
.75
.38
All dimensions in Inches
Tolerance
Eglin 1
Tetrahedral Frame
Part:
Center Main Base
UHMW-PE Disk
A: 0.25" Diameter Thru Hole
B: 0.99" Diameter Thru Hole
C: 0.47" Diameter Thru Hole
.01
.001
.001



Note: All dimension in inches.

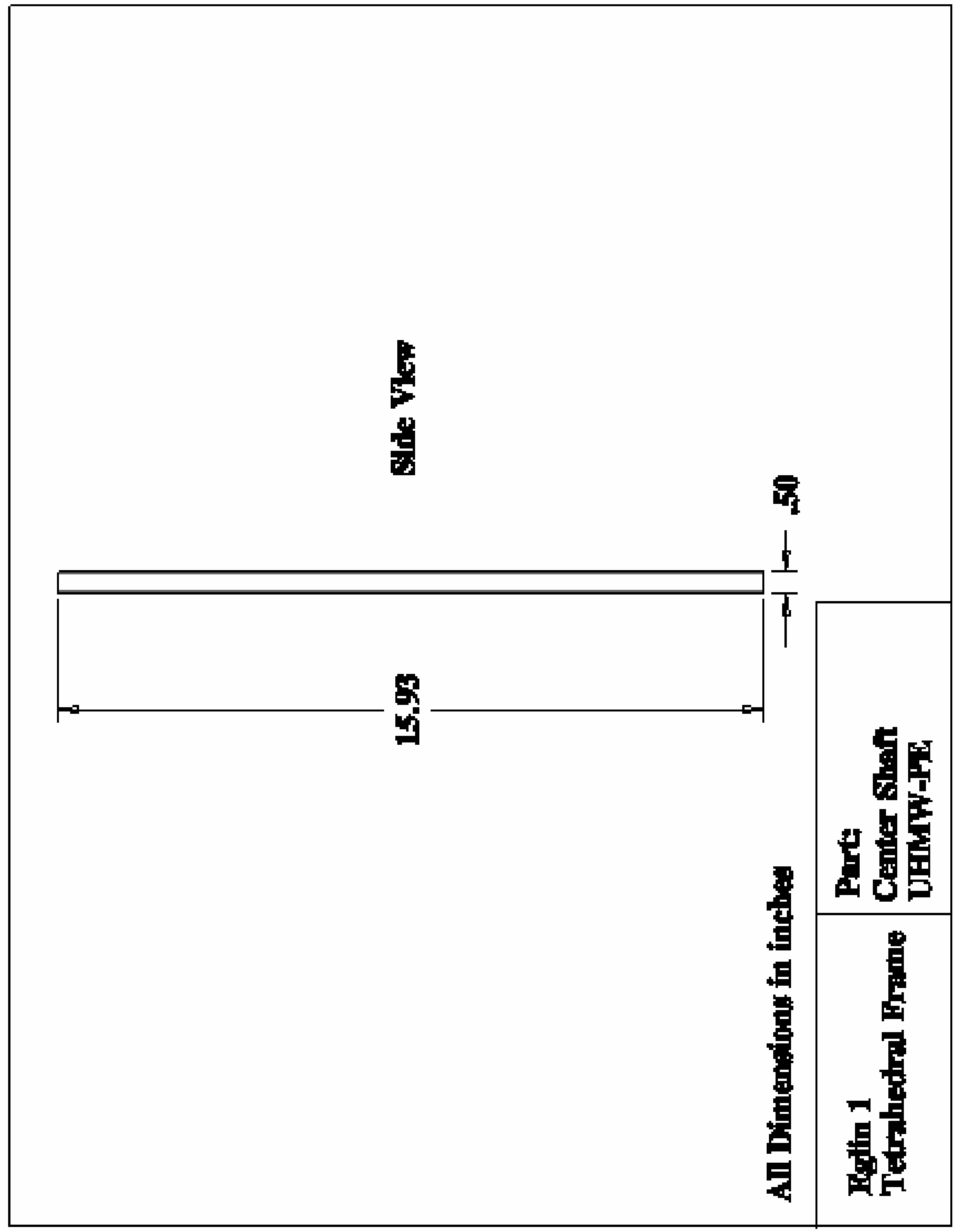
Eglin 1 Tetrahedral Frame	Part: Robot Adapter Plate 0.25" Aluminum Plate	A: Rounds are aesthetic only. B: 0.25 Inch Diameter Thru Hole C: 0.45 Inch Diameter Thru Hole

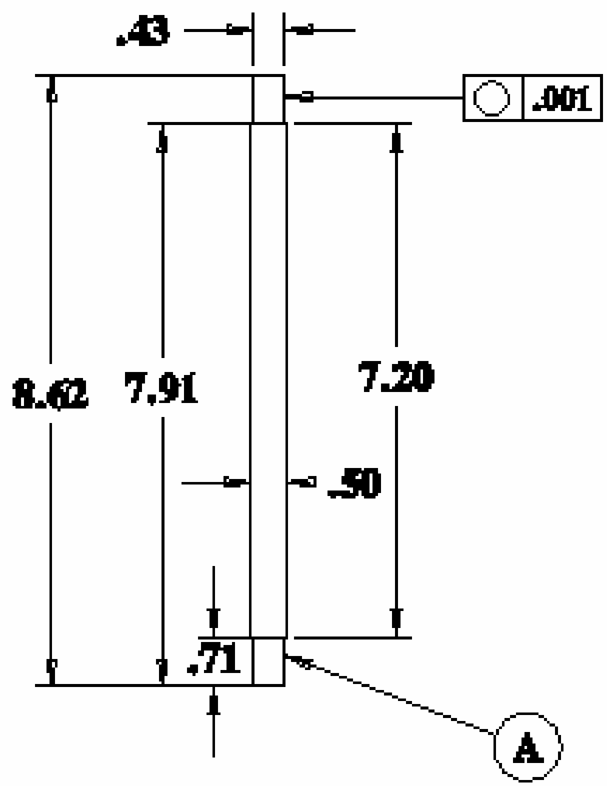
Tolerance	
⌖	.01
⌖	.01



All Dimensions in inches

Eglin 1 Tetrahedral Frame	Part: Connector Rod ABS Plastic	A: M12 x 1.25 Thread

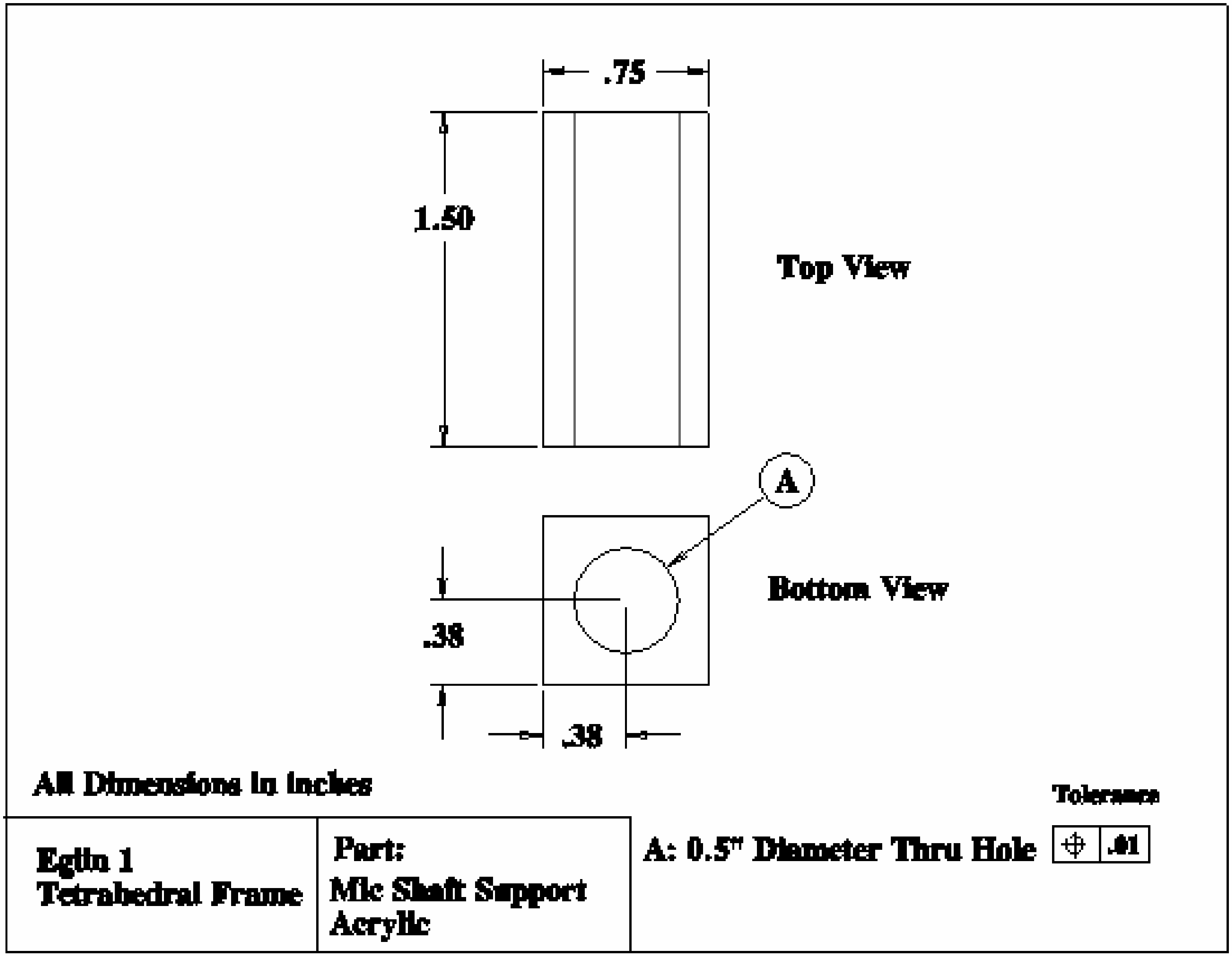

Side View
15.95
.50
All Dimensions in inches
Part:
Center Shaft
UHMW-PE
Eglin 1
Tetrahedral Frame

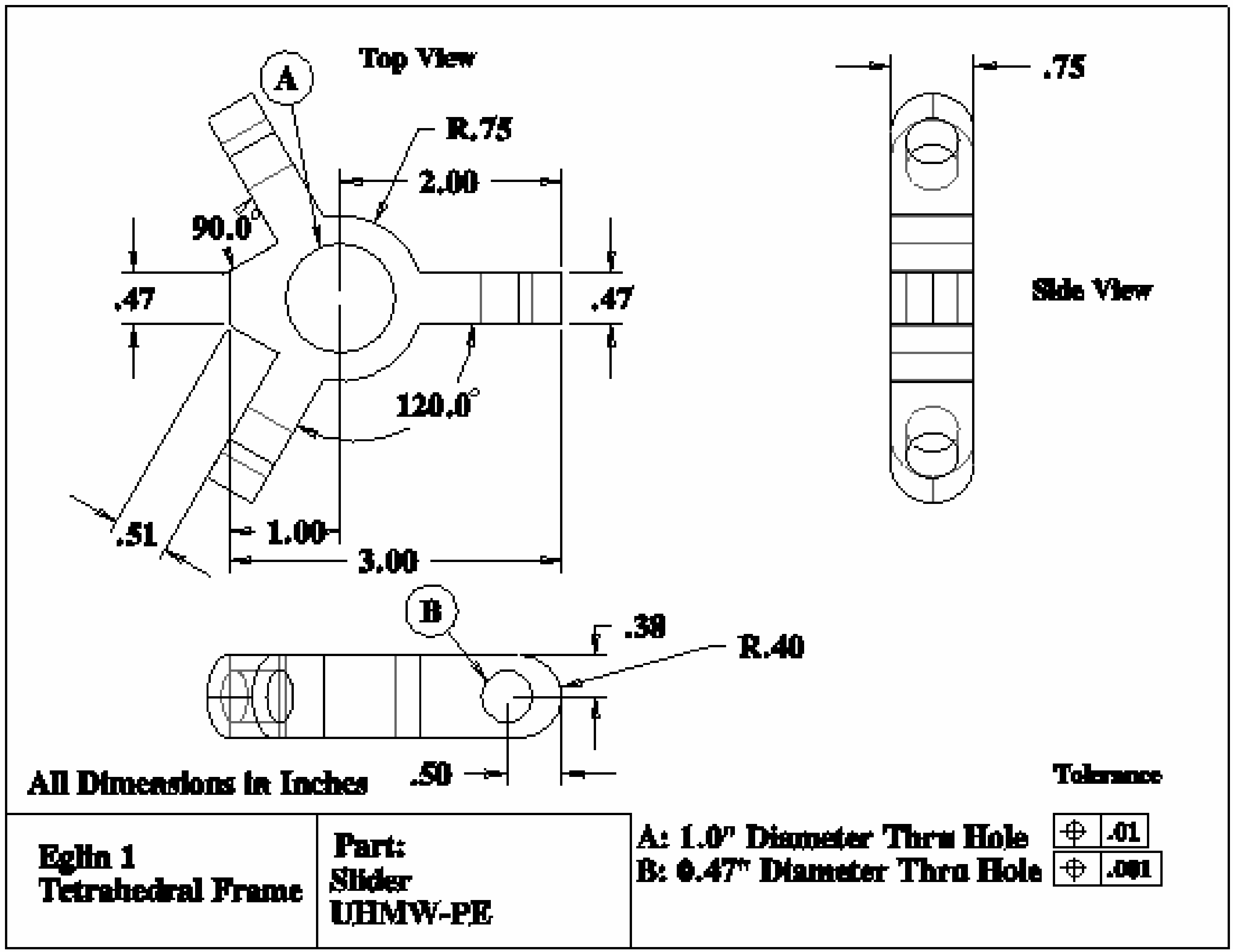


All Dimensions in inches

Eglin 1 Tetrahedral Frame	Part: Extension Rod ABS Plastic

A: M12 x 1.25 Thread


.75
1.50
Top View
A
Bottom View
.38
.38
All Dimensions In Inches
Tolerance
.01
Eglin 1
Tetrahedral Frame
Part:
Mic Shaft Support
Acrylic
A: 0.5" Diameter Thru Hole


Top View
A
R.75
2.00
90.0°
.47
.47
120.0°
.51
1.00
3.00
.75
Side View
B
.38
R.40
.50
All Dimensions in Inches
Tolerance
Eglin 1
Tetrahedral Frame
Part:
Slider
UHMW-PE
A: 1.0" Diameter Thru Hole
B: 0.47" Diameter Thru Hole
.01
.001

Appendix C: Purchase Orders/Receipts

McMaster Carr® Purchase Orders

4 FLORIDA STATE UNIVERSITY
204 BELMONT RD.
TALLAHASSEE FL 32301

(CALLER) KEVIN GARVEY

YOUR PURCHASE ORDER NUMBER
1114KGARVEY
Today's Date: 11/14/05

MCMASTER-CARR
6100 FULTON INDUSTRIAL BLVD
ATLANTA GA 30336-2852
IF THERE ARE ANY QUESTIONS ABOUT THIS SHIPMENT CONTACT OUR SALES DEPARTMENT
(404)346-7000

PAGE 1 OF 1
MCM NUMBER
1528809-02

Warehouse Location	McMaster Carr Part Number	Fill Quantity	Item Description	Your Line	Your Order	This Shipment
6- 5-05 34-81	8587 K43	5 FT	ABS (ACRYLONITRILE-BUTADIENE-STYRENE) ROD 1/2" DIAMETER,BLACK * 1 HNC Unit Price: 1.46 FT Extended Amount 7.30	1	5 FT	5
		*****	Information about the rest of your order		*****	
	8705 K332		UHMW POLYETHYLENE HOLLOW ROD Shipped 2 FT today from this McMaster-Carr warehouse	2	2 FT	
		*****	Charges for this shipment Merchandise Amount: 7.30 Total: $7.30		*****	

McMASTER-CARR PACKING LIST

900

16 FLORIDA STATE UNIVERSITY
204 BELMONT RD.
TALLAHASSEE FL 32301

(CALLER) KEVIN GARVEY

YOUR PURCHASE ORDER NUMBER
1114KGARVEY
Today's Date: 11/14/05

MCMASTER-CARR
6100 FULTON INDUSTRIAL BLVD
ATLANTA GA 30336-2852
IF THERE ARE ANY QUESTIONS ABOUT THIS SHIPMENT CONTACT OUR SALES DEPARTMENT
(404)346-7000

PAGE 1 OF 1
MCM NUMBER
1528809-01

Warehouse Location	McMaster Carr Part Number	Fill Quantity	Item Description	Your Line	Your Order	This Shipment
3-289-09 43-47	8705 K332	1 LG	UHMW POLYETHYLENE HOLLOW ROD 0.990" OD X 0.450" ID,2' LENGTH * 1 1 LG = 2FT Unit Price: 4.56 FT Extended Amount 9.12	2	2 FT	2
		*****	Information about the rest of your order		*****	
	8587 K43		ABS (ACRYLONITRILE-BUTADIENE-STYRENE) ROD Shipped 5 FT today from this McMaster-Carr warehouse	1	5 FT	
		*****	Charges for this shipment Merchandise Amount: 9.12 Shipping Charge: 13.75 Total: $22.87 Your shipping charge represents shipping charges for your entire order. You will not be charged for additional shipments.		*****	

Lowe’s® Receipts

LOWE'S

FORT WALTON BEACH, FL
(850)863-0900
-RETURN-
: S04797N4 778998   11-25-05

1/4"X1-1/8"X2-1/4   2.25-
ORE: 479 DATE:112505 INV:11856
3 @ 0.75-

SUBTOTAL: 2.25-
TAX 38470 :   0.14-
ICE 36402 TOTAL: 2.39-

BALANCE DUE: 2.39-

CASH :   2.39-

MINAL: 36 11/25/05 15:15:17

LOWE'S

FORT WALTON BEACH, FL
(850)863-0900
-SALE-
SALES #: S0479CH2 16877   11-25-05

137266 SET SCRW 1/4-20X5   1.36
2 @ 0.68

SUBTOTAL: 1.36
TAX 38470 :   0.09
INVOICE 14148 TOTAL: 1.45

BALANCE DUE: 1.45

CASH :   5.00
CHANGE :   3.55

0479 TERMINAL: 14 11/25/05 11:08:30

LOWE'S

FORT WALTON BEACH, FL
(850)863-0900
-SALE-
SALES #: S04790K2 916905   11-25-05

136606 1/4X1 1/2 RUBB   1.84
2 @ 0.92
136082 FLAT WASHERS 1/4-   1.04
136012 HEX HD BOLTS 1/4-   2.30
2 @ 1.15
136006 HEXNUTS 1/4-20 BR   1.04
136978 1/2X. 257X1/4 NYL   1.84
2 @ 0.92
19666 1/4"X1-1/8"X2-1/4 [illegible]
3 @ 0.75
135999 LOCK WASHERS 1/4   1.04

SUBTOTAL: 11.35
TAX 38470 :   0.69

Appendix D: Component Specifications

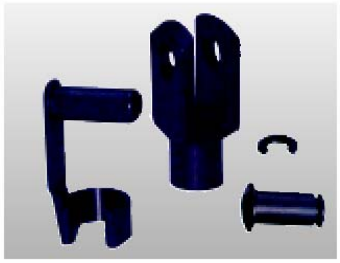
igubal® Clevis Joint

Telephone 1-888-803-1895
Fax 1-401-438-7680

Internet: http://www.igus.com
E-Mail: webmaster@igus.com
QuickSpec: www.igus.com/qs/igubal.asp

30.2

igus® igubal® Clevis Joint



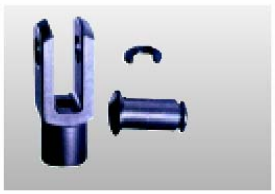
The most significant Properties

- lightweight
- universal corrosion resistance
- high tensile strength
- can be used in combination with rod end bearings of the dimensional series E
- vibration dampening
- noise dampening
- available in left and right threads

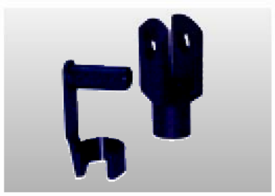
igubal® clevis joints are all made of igumid G according to DIN 71752, which can be used in combination with the rod end bearings of the dimensional series E. Available components are clevis joint, clevis pin and clip or as an alternative, spring-loaded pin.

Part Selections

All of the components listed below are also available individually



Clevis joint with clevis pin and clip
GER(L)MK-



Clevis joint with spring-loaded pin
GER(L)MF-

Clevis joint with clevis pin and clip and rod end bearing
GER(L)MKE-

Clevis joint with spring loaded pin and rod end bearing
GER(L)MFE-

Structure of the part numbers for igubal® Clevis Joints

The part numbers of igubal clevis joints are designed according to the following system:

G E R M KE - 12 F

Model
G = Clevis Joint

Dimensional Series
E = dimensional series E

Thread design
L = left
R = right

Measuring system
M = metric

Options
None Standard
K = with clevis pin and clip
KE = with clevis pin, clip, and rod end bearing
F = with spring-loaded hasp pin
FE = with spring-loaded hasp pin and rod end bearing

Nominal dimension of inside diameter

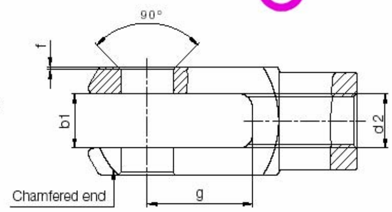
Design
None standard thread
F = Fine thread

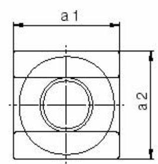
The example shows a complete clevis joint assembly of the dimensional series E with inner right threading in metric dimensions. The inner diameter of the spherical ball is 12mm. The thread bore has a metric fine threading (M12 x 1.25).

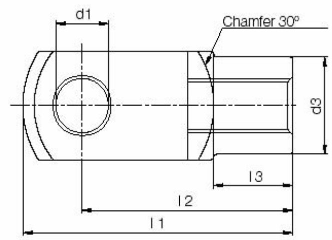
Source: Igus®. 2005©. webmaster@igus.com. 12 November 2005 through 1 December 2005. <www.igus.com>

igubal® Clevis Joint - mm - GERI / GELI GERIK/GELIK

- lightweight
- universal corrosion resistance
- high tensile strength
- Can be used in combination with rod end bearings of the dimensional series E
- vibration dampening
- noise dampening
- available in left and right thread







Dimensions (mm)

Right (Left) Thread	d1	g	a1	a2	b1	d2	d3	f	l1	l2	l3
	H9	h11		+0.3 -0.16	B13	Thread-Tolerance 6H	+0.3 -0.3	+0.3 -0.3	+0.5 -0.5	+0.3 -0.3	+0.2 -0.
GER(L)I-03	5	10	10	10	5	M04	9	0.5	26.0	20.0	7.5
GER(L)I-04	5	12	12	12	6	M05	10.0	0.5	30.6	24.0	9.0
GER(L)I-05	8	16	16	16	8	M08	14.0	0.5	41.6	32.0	12.0
GER(L)I-06	10	20	20	20	10	M10	18.0	0.5	51.3	40.0	15.0
GER(L)I-07	12	24	24	24	12	M12	20.0	0.5	61.3	48.0	18.0
GER(L)I-08	15	28	27	27	14	M14	24.0	0.5	71.3	56.0	22.5

Load Data

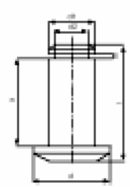
Right (Left) Thread	Max. Static Axial Tensile Strength GERM		Right (Left) Thread	Max. Static Axial Tensile Strength GERMK	
	Short-term (lbs)	Long-term (lbs)		Short-term (lbs)	Long-term (lbs)
GER(L)I-03	225	112	GER(L)IK-03	202	90
GER(L)I-04	270	135	GER(L)IK-04	202	101
GER(L)I-05	607	303	GER(L)IK-05	472	236
GER(L)I-06	1056	528	GER(L)IK-06	674	404
GER(L)I-07	1281	640	GER(L)IK-07	787	393
GER(L)I-08	719	360	GER(L)IK-08	629	315

➤ Tolerance Table, Page 1.24

igubal® Clevis Pin - inch/mm - GBI

Part Number Pin	d1	d2	d	l	l1	m	Clip
GBI-03	.1875	.1260	.3125	.55	.3975	.0472	GSR-04
GBI-04	.2500	.1969	.3750	.65	.4764	.0512	GSR-08
GBI-05	.3125	.1969	.4375	.85	.6339	.0512	GSR-08
GBI-06	.3750	.2756	.5000	1.05	.7953	.0591	GSR-10
GBI-07	.4375	.3543	.6250	1.25	.9528	.0669	GSR-12
GBI-08	.5000	.3543	.7500	1.40	1.0709	.0669	GSR-12

Part Number Pin	d1	d2	d	l	l1	m	Clip
GBM-03	5	3.2	8	14	10	1.20	GSR-04
GBM-04	6	5.0	10	17	12	1.30	GSR-08
GBM-05	8	5.0	11	22	16	1.30	GSR-08
GBM-06	10	7.0	13	27	20	1.50	GSR-10
GBM-07	11	9.0	16	32	24	1.70	GSR-12
GBM-08	13	9.0	19	36	27	1.70	GSR-12



igubal® Clevis Joint

GERM / GELM

PDF: www.igus.com/pdf/igubal.asp
Specs/CAD/Samples/Size Selector/RFQ:
www.igus.com/igubal.asp

inch

mm

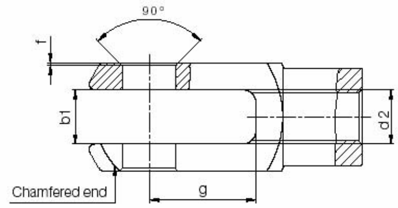
Source: Igus®. 2005©. webmaster@igus.com. 12 November 2005 through 1 December 2005. <www.igus.com>

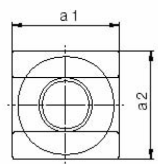
Telephone 1-888-803-1895
Fax 1-401-438-7680

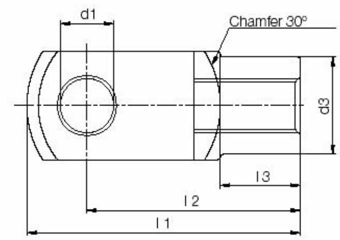
Internet: http://www.igus.com
E-Mail: webmaster@igus.com
QuickSpec: www.igus.com/qs/igubal.asp

igubal® Clevis Joint - mm - GERM / GELM GERMK/GELMK

- lightweight
- universal corrosion resistance
- high tensile strength
- Can be used in combination with rod end bearings of the dimensional series E
- vibration dampening
- noise dampening
- available in left and right thread







Load Data

Right (Left) Thread	Max. Static Axial Tensile Strength GERM Short-term (lbs)	Long-term (lbs)	Right (Left) Thread	Max. Static Axial Tensile Strength GERMK Short-term (lbs)	Long-term (lbs)
GER(L)M-04	179	90	GER(L)MK-04	135	67
GER(L)M-05 DIN M4	225	112	GER(L)MK-05 DIN M4	202	90
GER(L)M-05 DIN M5	225	112	GER(L)MK-05 DIN M5	180	90
GER(L)M-05	270	135	GER(L)MK-05	202	101
GER(L)M-06	314	157	GER(L)MK-06	292	146
GER(L)M-08	607	303	GER(L)MK-08	472	236
GER(L)M-10	1056	528	GER(L)MK-10	674	404
GER(L)M-10 F	1056	528	GER(L)MK-10 F	674	404
GER(L)M-12	1281	640	GER(L)MK-12	787	393
GER(L)M-12 F	1281	640	GER(L)MK-12 F	787	393
GER(L)M-14	1483	741	GER(L)MK-14	1371	685
GER(L)M-15	719	360	GER(L)MK-15	629	315
GER(L)M-16	1686	843	GER(L)MK-16	1573	786
GER(L)M-16 F	1686	843	GER(L)MK-16 F	1573	786
GER(L)M-20	2136	1068	GER(L)MK-20	2023	1012
GER(L)M-20	2136	1068	GER(L)MK-20	2023	1012

Dimensions (mm)

Right (Left) Thread	d1 H9	g h11	a1	a2 +0.3 -0.16	b1 B13	d2 Thread-Tolerance 6H	d3 +0.3 -0.3	f +0.3 -0.3	l1 +0.5 -0.5	l2 +0.3 -0.3	l3 +0.2 -0.
GER(L)M-04	4	8	8	8	4	M4	8.0	0.5	21.0	20	7.5
GER(L)M-05 DIN M4	5	10	10	10	5	M04	9	0.5	25.5	20	7.5
GER(L)M-05 DIN M5	5	10	10	10	5	M04	9	0.5	25.5	20	7.5
GER(L)M-05	5	12	12	12	6	M05	10.0	0.5	30.6	24.0	9.0
GER(L)M-06	6	12	12	12	6	M06	10.0	0.5	30.6	24.0	9.0
GER(L)M-08	8	16	16	16	8	M08	14.0	0.5	41.6	32.0	12.0
GER(L)M-10	10	20	20	20	10	M10	18.0	0.5	51.3	40.0	15.0
GER(L)M-10 F	10	20	20	20	10	M10x1.25	18.0	0.5	51.3	40.0	15.0
GER(L)M-12	12	24	24	24	12	M12	20.0	0.5	61.3	48.0	18.0
GER(L)M-12 F	12	24	24	24	12	M12x1.25	20.0	0.5	61.3	48.0	18.0
GER(L)M-14	14	28	27	27	14	M14	24.0	0.5	71.3	56.0	22.5
GER(L)M-15	15	28	27	27	14	M14	24.0	0.5	71.3	56.0	22.5
GER(L)M-16	16	32	32	32	16	M16	26.0	1.0	81.9	64.0	24.0
GER(L)M-16 F	16	32	32	32	16	M16x1.5	26.0	1.0	81.9	64.0	24.0
GER(L)M-20	20	40	40	40	20	M20x1.5	34.0	1.0	105.0	80.0	30.0
GER(L)M-20	20	40	40	40	20	M20x2.5	34.0	1.0	105.0	80.0	30.0

Imperial sizes available. Minimum quantities may be required.

➤ Tolerance Table, Page 1.24

Source: Igus®. 2005©. webmaster@igus.com. 12 November 2005 through 1 December 2005. <www.igus.com>

SORBOTHANE® Standard Products & Price Guide

Sorbothane, Inc. / 2144 State Route 59 / Kent, Ohio 44240 / USA
www.sorbothane.com / ph 800.838.3906 / fax 330.678.1303
effective 1 April 2005

INTRODUCTION

Sorbothane® is a proprietary, visco-elastic polymer. Visco-elastic means that a material exhibits properties of both liquids (viscous solutions) and solids (elastic materials).

Sorbothane is a proprietary thermoset, polyether-based, polyurethane material.

Sorbothane combines shock absorption, good memory, vibration isolation and vibration damping characteristics. In addition, Sorbothane is a very effective acoustic damper and absorber. While many materials exhibit one of these characteristics, Sorbothane combines all of them in a stable material with a long fatigue life.

- Sorbothane has a low creep rate compared to other polymers (rubber, neoprene, silicone, etc.)
- Sorbothane has a superior damping coefficient, over a very wide temperature range, compared to any other polymer.
- Unlike fluid-based shock absorbers or foam products, Sorbothane absorbs shock efficiently for millions of cycles.
- Sorbothane eliminates the need for metal springs to return the system to its equilibrium position after absorbing a shock.

CUSTOM SOLUTIONS

If an appropriate isolator cannot be found among the standard products, Sorbothane also manufactures a host of custom isolators. Additionally, you may make isolators yourself using standard sheet stock. See the stock sheet section or contact the factory for further information.

ELECTRONICS

This catalog shows "industrial-sized" isolators for loads in pounds, hundreds of pounds and thousands of pounds. Sorbothane is widely used to isolate delicate electronics – printed-circuit boards, LCD's, disk drives and other small devices. Space is always tight in these applications.

Applications range from rocket launchers to cell phones. Volumes range from 50 pieces to 100,000's of pieces. Consult the factory for these specials.

SHOCK ABSORBERS

In vibration, the wrong design can make matters worse than before. In shock absorption, even a small amount of Sorbothane can produce results. Consult the factory on your shock applications.

ACOUSTICS

The Sorbothane hemispheres are world-famous mounts for audio equipment. In addition, where space is tight, thin sheets of Sorbothane, with or without Pressure Sensitive Adhesive (PSA) can convert a noisy design into a very quiet success.

GASKETS

Sorbothane is a popular material for gaskets because of its chemical resistance, its ability to conform to irregular surfaces, its low creep and its reusability. Its natural tackiness makes it easy to install. Gaskets can be made by you from sheet stock or produced at our factory.

PROCEDURE FOR SELECTING A VIBRATION ISOLATOR

1. Determine the load (weight) of the unit that requires isolation. Divide the weight of the system by the number of isolators to be used. This is the load per isolator.

Source: Sorbothane®. 15 September 2005©. Onesixtyeight internet studios, llc. 15 September 2005 through 1 December 2005. <www.sorbothane.com>

2. Determine the lowest critical frequency (**excitation frequency**) of the system. For example: RPM of the motor, rate at which a cylinder strokes, output speed of a speed reducer. Divide minute-based information, such as RPM, by 60 to convert to Hertz (cycles per second).

3. Select an appropriate isolator by using the information charts supplied with each product. Generally, an appropriate isolator will create a **system natural frequency** at least one-third lower than the excitation frequency. For instance, if the source of the vibration is an 1800 RPM motor, then the excitation frequency is 1800/60 = 30 Hertz. The resultant, (damped) system natural frequency should be 20 Hertz or less.

4. Select the isolator that meets the technical and physical requirements of your system. In addition to a part number, you must also specify the durometer of the Sorbothane part required to get the specified load rating. **Do not oversize isolators**. Oversized isolators raise the natural frequency and reduce the effectiveness. Sorbothane's standard colors are black, gray and royal blue. Other colors are available for large quantity special orders.

5. Remember: A bad design can actually make things worse for vibration damping. Consult the factory if you are having problems.

6. Remember: Sorbothane works best in compression. If you have a tension application, bushings or a custom variation of the bushings are your best options.

DUROMETER

Durometer is a measure of relative stiffness and is used to compare polymers.

Sorbothane is softer than rubber and most other polymers. Sorbothane is measured on the Shore "00" scale.

Most types of rubber and other polymers are specified using the Shore "A" or Shore "D" scales. In comparing stiffness, be aware of the scale being used for the material in question.

The softest Sorbothane, 30 durometer, is used for high frequency (15 kiloHertz and up) and low temperature (less than - 20°F.). It has been described as having the consistency of wet chewing gum but with memory of its as-cast shape.

Use 70 durometer Sorbothane for high performance isolation for low frequencies (less than 500 Hertz) or where material toughness is paramount.

50 durometer Sorbothane is a compromise in strength and isolation.

Intermediate durometers may be cast for special applications as required.

The stiffness comparison values listed are approximate and should only be used as a guide.

Shore Hardness Scale Comparison Chart

A	B	C	D	0	00
100	85	77	58		
95	81	70	46		
90	76	59	39		
85	71	52	33		
80	66	47	29	84	98
75	62	42	25	79	97
70	56	37	22	75	95
65	51	32	19	72	94
60	47	28	16	69	93
55	42	24	14	65	91
50	37	20	12	61	90
45	32	17	10	57	88
40	27	14	8	53	86
35	22	12	7	48	83
30	17	9	6	42	80
25	12			35	76
20	6			28	70
15				21	62
10				14	55
5				8	45

Source: Sorbothane®. 15 September 2005©. Onesixtyeight internet studios, llc. 15 September 2005 through 1 December 2005. <www.sorbothane.com>

Bushings

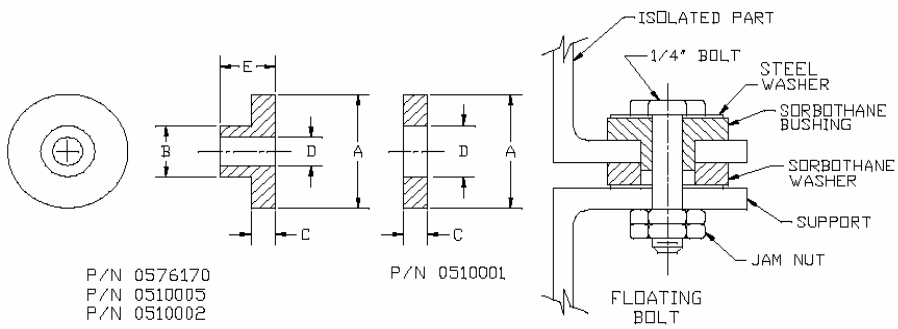
Sorbothane bushings can isolate vibration and absorb shock.

Using two bushings together or a bushing and washer(s), you can create a floating bolt that isolates the unit from any metal-to-metal contact.

Different durometers allow for different spring rates. Load ratings assume a 10%-20% deflection of the material. This deflection is achieved by a combination of system weight and torquing of the connecting bolts.

Do not over torque the bolts as this defeats the intended vibration isolation and shortens bushing life. Use a high-quality (thread deforming) lock nut or doubled jam nuts to prevent vibration from loosening bolt.

Unlike other Sorbothane products, bushings may be used in tension connections.



	Dimensions (Inches)					Load Rate (lbs)		
Part Number	A	B	C	D	E	30 Duro	50 Duro	70 Duro
0576170	0.72	0.50	0.15	0.28	0.22	3-8	4-11	8-21
0510005	1.00	0.46	0.19	0.28	0.31	7-18	22-35	20-42
0510001	1.00	-	0.19	0.45	-	3-4	5-7	8-11
0510002	1.00	0.45	0.19	0.25	0.60	13-22	22-33	28-44

	Unit Price in USA Dollars for Purchase Quantity Ranges					
Part Number	100 to 249	250 to 499	500 to 999	1000 to 4999	5000 to 9999	10,000+
0576170	1.06	0.61	0.48	0.37	0.35	0.31
0510005	1.08	0.66	0.51	0.43	0.37	0.36
0510001	1.08	0.66	0.51	0.43	0.37	0.36
0510002	1.22	0.85	0.69	0.57	0.51	0.48

To order: Specify part number, durometer, quantity, color. Color black is standard. Other colors and dimensional variations are available by special order.

Source: Sorbothane®. 15 September 2005©. Onesixtyeight internet studios, llc. 15 September 2005 through 1 December 2005. <www.sorbothane.com>

Appendix E: Background Research

Vibration Damping

TECHNICAL
PRODUCT
FUNCTION
SHEET

Foamex

Acoustical Functions

Absorbing · Dampening · Acoustical Transparency

General Description

Foamex acoustical materials are designed to perform a number of specific and seemingly opposite acoustical functions…either reducing sound levels, or passing sound undistorted and undiminished…each determined by both the basic design of the foam and its method of application. Certain non-reticulated foams are ideal for sound absorption and attenuation. Reticulated foams can absorb sound very effectively or can be completely acoustically transparent. Noise absorption applications include: commercial aircraft ventilation ducts; headliners and back panels for tractor cabs and off-road vehicles; data-processing equipment; portable air compressors and power units; appliances; snowmobiles; as well as headliners and panels for automobiles.

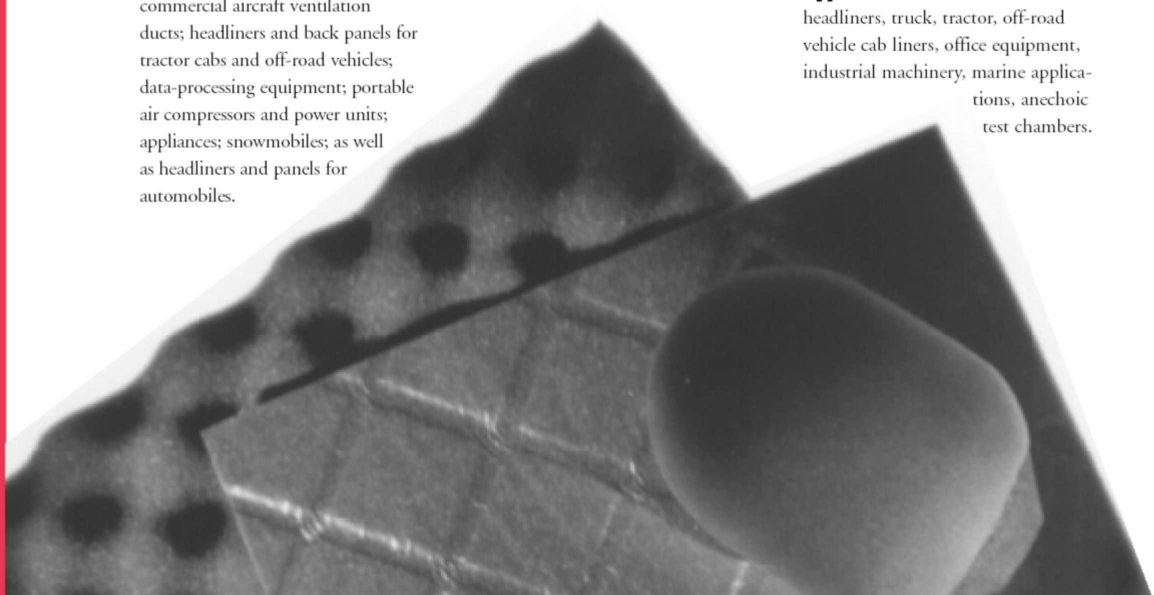
Sound fidelity applications include: stereo speaker grilles, earphones, and microphone covers.

Benefits included are: predictable sound absorption in broad band (low, mid, high) frequency range from engineered foam grades; nearly "total perfection in sound transparency applications (polyurethane foam is 97% air) from other selected grades; fabrication design flexibility; functional/decorative laminate capabilities; installation ease; excellent shape retention, and resistance to wear and abrasion.

Noise Absorption

Acoustical foam will absorb the activity of airborne or fluid-borne noise, causing a loss in energy by weakening reflected vibrations. The sound vibration pulses are literally "tired out" by their effort to force the foam strands to vibrate…and the noise reflected level is, therefore, immediately reduced. Acoustical foam for sound absorption provides the foam high efficiency, consistent throughout, and predictable from installation to installation.

Applications Include: Automobile headliners, truck, tractor, off-road vehicle cab liners, office equipment, industrial machinery, marine applications, anechoic test chambers.



Source: <u>Foamex Shaping things to come.</u> 2004©. Foamex LP. 1 October 2005 through 1 December 2005. <www.foamex.com>

Acoustical Functions

Product	Function	Application
*SIF® (Fine and Coarse Pore)	Sound Absorber	Anechoic chambers and noise test facilities
*SIF® (Medium Pore)	Air Diffuser	Microphone wind screen covers, power brake units, mufflers
*SIF II® (Medium Pore)	Acoustically Transparent	Hi Fi speaker grilles
*SIF Felt®	Sound Absorber	Tuned for mid and high frequencies
*Aerofonic®	Sound Absorber	FAR 25.853 extended environmental use felt designed for aircraft
**Aresto II®	Sound Absorber	General use where flammability and economy are important
**Pyrell®	Sound Absorber	General use where maximum flammability protection is a concern
HyFonic®	Sound Absorber	An extended environmental foam where flammability is a concern
Fine-Pore Acoustical	Sound Absorber	General use where economics is the major concern
Custom Lamination	Sound Absorber Moisture Barrier	Where special barriers are needed and/or aesthetics are important

**Engineered to be predictable from run to run based on close tolerances, for density permeability and pores per lineal inch (ppi)*

***Available in 4 lb/ft³ density*

Typical Physical Properties of Foamex Acoustical Materials

	SIF® Industrial Foam	SIF Felt®	Aresto® II	Pyrell®	HyFonic® I	Aerofonic® Felt
Grade	90 ppi	3-900	70 ppi	70 ppi	65 ppi	4-700
Density (lb/cu. ft)	1.9	n/a	2.0	2.0	1.7	n/a
Tensile Strength (PSI)	35	n/a	20.0	22	14	n/a
Elongation (%)	415	n/a	190	220		n/a
Polyol	Polyester Polyurethane	Polyester Polyurethane	Polyester Polyurethane	Polyester Polyurethane	Polyether Polyurethane	Polyether Polyurethane

Tested in accordance with ASTM 3574. Physical properties not to be used as a specification.

Source: Foamex Shaping things to come. 2004©. Foamex LP. 1 October 2005 through 1 December 2005. <www.foamex.com>

Vibration Isolation

Noise transmission sound barriers are used to reduce the noise level being transmitted through a housing when the housing itself does not satisfactorily perform that function. When noise levels are severe, an additional high mass transmission-reduction wall or septum is often required. Acoustical foam is applied as a decoupler between the housing and the inner septum, and as an absorber on the outer septum wall, to effectively absorb and reduce the noise energy flow.

Applications Include: Acoustical panels, aircraft, turbine engines, broadcast studios' data processing equipment, power generator housings, automotive headliner.

Vibration Damping

Acoustical foams are used to reduce vibrations of physical structures that, in turn, produce noise due to that vibration. In some cases, for maximum vibration reduction, acoustical foams are used in conjunction with a damping layer, such as a viscoelastic material.

Applications Include: Air conditioning equipment, dishwashers, aircraft compartments, high-speed rail cars, data processing machines, enclosed power units, engine housing.

Sound Fidelity

Reticulated acoustical foams have been proven virtually acoustically transparent; effectively invisible to sound waves in audible frequencies. This is true even of foam two inches thick. A leading California stereo speaker manufacturer made this fact well-known by introducing a line of speakers that offered nearly perfect sound transparency through striking design-sculptured foam grilles in a variety of colors.

Applications Include: Stereo speaker grilles, earphones, microphone windscreen covers, smoke alarm grilles.

Acoustical Foam Types
Industrial Foam (SIF®)

SIF® is reticulated, 90 pores-per-linear-inch (ppi) polyurethane foam with excellent uniformity and predictability. It is particularly effective where it can be used in a thickness above two inches. . .in the anechoic chambers of sound testing facilities, for example.

SIF Felt® Acoustical Foam

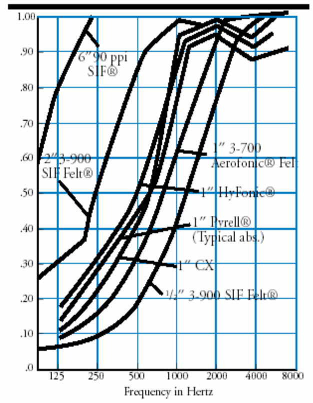
SIF Felt® is 90 ppi reticulated foam that has been permanently compressed by both heat and pressure; sound absorption properties can be tuned to selected frequencies by specifying thickness and final felt "firmness grade." SIF Felt® is specified where both space and predictable performance are important factors.

Custom Lamination

Perforated vinyl, Mylar* adhesive, or other approved substrates, can be specifically flame-bonded to Foamex acoustical materials according to specific customer needs.

*Registered trademark of The DuPont Company.

Typical Values of Random Absorption Coefficients

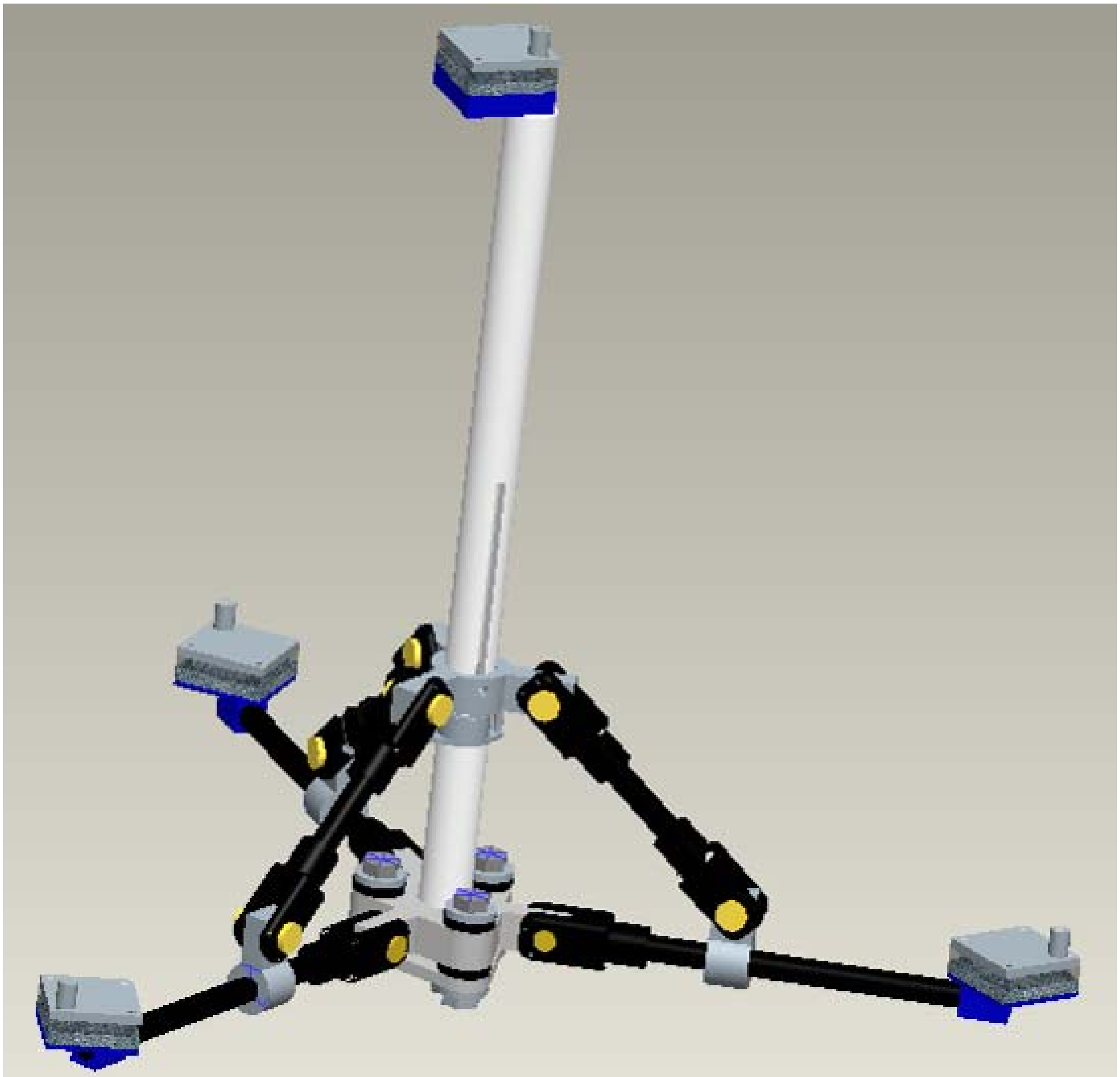


Source: Foamex Shaping things to come. 2004©. Foamex LP. 1 October 2005 through 1 December 2005. <www.foamex.com>

Tetrahedral Collapsible Array Operations Manual

Tetrahedral Collapsible Array
Installation and Operating Manual

Eglin 1 Senior Design Team



Tetrahedral Collapsible Array Housing

The tetrahedral array frame has been designed and built to house four microphones in a tetrahedral orientation with a microphone-to-microphone distance of twenty inches.

Intended Use

Tetrahedral Array frame integrates acoustic sensor eye to RDS (Robot Demonstration System) or any other applicable robot platform. Operation of array damps mechanical vibration received through the structure to the acoustic eye sensor caused by the movement of the robot platform.

Function

Physically attaches acoustic sensor eye to applicable platform. Array frame decouples acoustic sensor eye from direct contact with any applicable platform. Remotely retracts or deploys acoustic sensor eye for transportation or navigational needs.

⚠ Safety Instructions ⚠

Incorrect wire connection will result in stepper motor damage or possible fire hazard. **DO NOT** cross coil wiring on stepper motor (reference stepper motor controller manual for proper wiring schematics).

Keep extremities clear from array at all times during retraction and deployment.

Stepper motor may become hot during operation (Temperatures up to 75°C). **DO NOT** touch stepper motor during operation and up to 15 minutes after operation.

Keep fingers away from guide track (slider slot) at all times.

Tetrahedral Array Precautions

DO NOT operate stepper motor while tetrahedral array is in the toggled position. Severe damage may occur to lead screw and or stepper motor.

DO NOT continue to operate stepper motor once fully deployed or retracted position is reached. Stepper motor damage may occur.

DO NOT allow array to come in close proximity with temperatures in excess of 120° F.

DO NOT over torque the Sorbothane® bushing bolts. Do not exceed 5 in-lbs of torque.

DO NOT over tighten setscrews. Stripping of thread material **WILL** occur.

DO NOT impose heavy loads on array frame. Material damage **WILL** occur.

DO NOT hand drive the slider mechanism. Binding may occur in array frame.

Features

Actuated by a single bi-polar stepper motor.

Utilizes simple four-bar slider linkages for retraction and deployment.

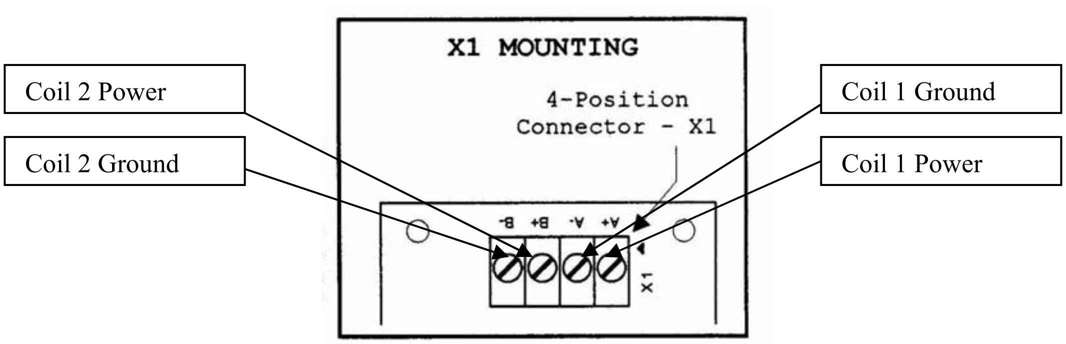
Integrates a sliding, dual slider, slot track system for actuation.

Full collapsibility into toggled position for ease of transportation.

Complete vibration damping characteristics enhanced throughout the array frame.

Adjustability of the microphone-to-microphone distances on the apices of the tetrahedral frame.

Adaptability of multiple microphone sensor designs utilizing optional adapter plates.



(Figure 1)

Assembly

Major components of tetrahedral array come pre-assembled. Only small adjustments of rods are required to operate array.

Adjustments of rods can be viewed in the exploded assembly drawings in the assembly manual.

Connect stepper motor to stepper motor controller following wiring sequence seen in figure 1. Reference stepper motor controller manual for further schematics.

If complete disassembly and reassembly is required see schematic layout in the assembly manual.

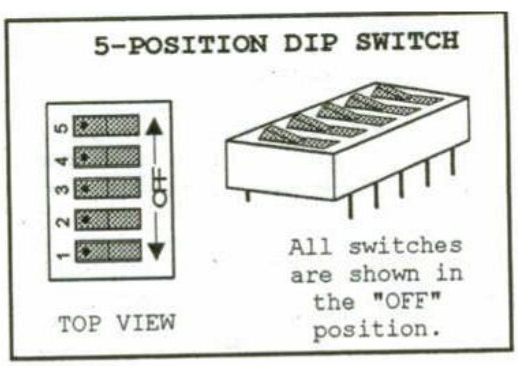
Operation

To deploy or retract tetrahedral array frame follow the following sequence.

1.) Adjust switch 3 on stepper motor controller to change direction of stepper motor for deployment or retraction.
    a. For deployment place switch 3 in the open position as seen in figure 2.
    b. For retraction place switch 3 in the closed position as seen in figure 2.

2.) Adjust switch 2 on stepper motor controller to change between full or half step mode. (For explanation of full or half step mode see stepper motor controller manual).
    a. For Full Step place switch 2 in close position as seen in figure 2.
    b. For half Step place switch 2 in open position as seen in figure 2.

3.) Power on stepper motor by placing switch 1 in the open position as seen in figure 2.



(Figure 2)

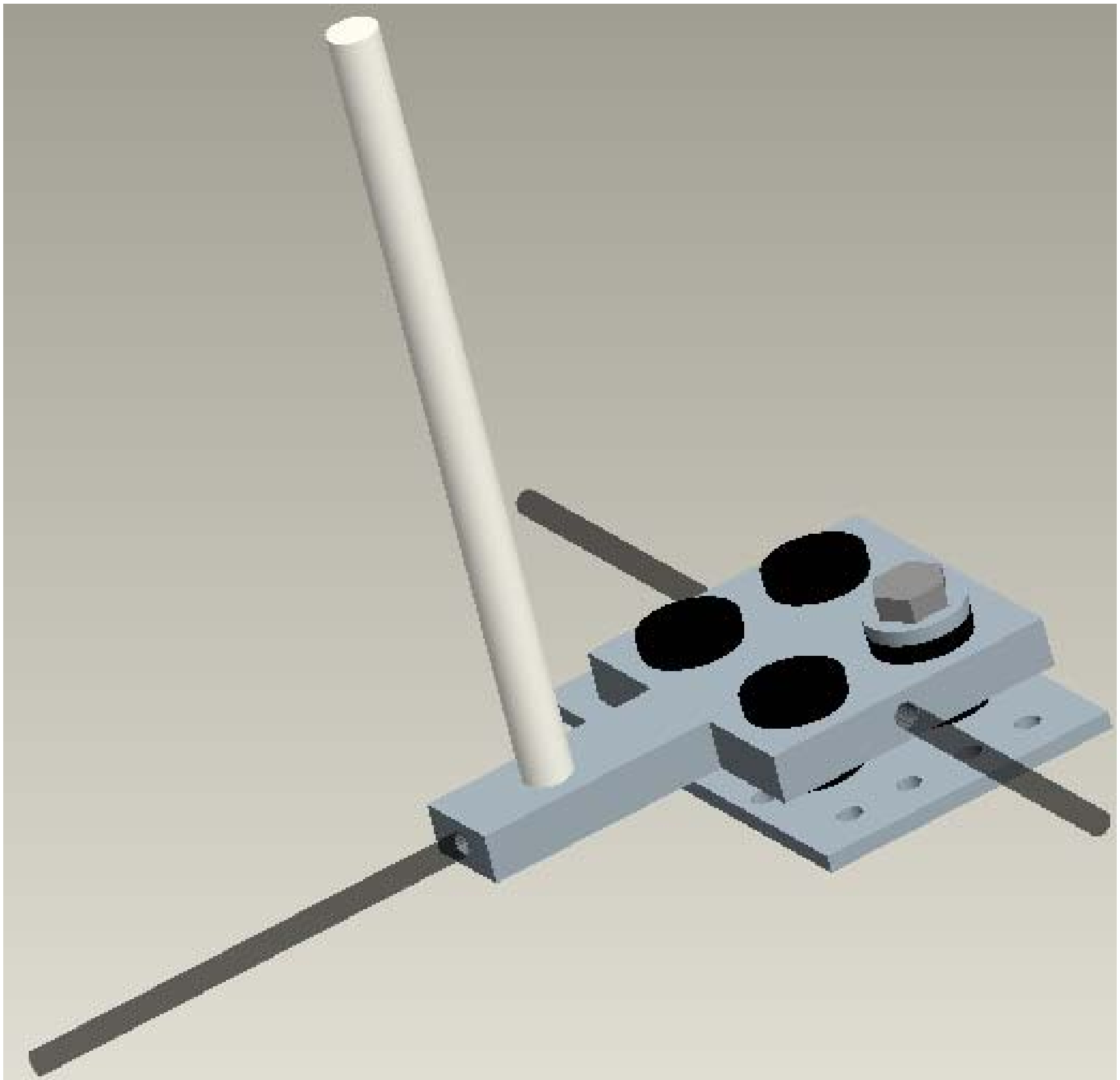
Maintenance/ Adjustment

Tetrahedral array comes preassembled with main adjustments already in place. For fine-tuning of microphone sensors all connecting rods, extension rods, microphone mounts and adapter plates are adjustable using a 5/64" Allen wrench.

For maintenance purposes follow these key points:

1.) Lubricate slider track periodically if motor becomes labored.
2.) Lubricate pin joints if binding occurs.
3.) Clean any dirt or debris from slider track before operation.
4.) Check periodically for signs of wear and tear and replace any damaged or worn components.
5.) Check floating bolt torque to ensure complete damping characteristics on array.

T-Base Array
Installation and Operating Manual
Eglin 1 Senior Design Team



T-Base Array Housing

The T-Base array that has been designed and built is a frame to hold four microphones in a tetrahedral orientation with a microphone to microphone distance of ten inches. This array is a non-collapsible array in a "T" orientation. This array features the same vibration isolation mechanisms as the collapsible array.

Intended Use

T-Base Array integrates acoustic sensor eye to VEX™ Robot. Operation of array damps mechanical vibration received through the structure to the acoustic eye sensor caused by the movement of the robot platform.

Function

Physically attaches acoustic sensor eye to applicable platform. Array frame decouples acoustic sensor eye from direct contact with any applicable platform.

⚠ Safety Instructions ⚠

T-Base Array contains protruding edges that may cause harm to user if not carefully handled.

⚠ T-Base Array Precautions ⚠

DO NOT allow array to come in close proximity with temperatures in excess of 120° F.

DO NOT over torque the Sorbothane® bushing bolts. Do not exceed 5 in-lbs of torque.

DO NOT over tighten setscrews. Stripping of thread material **WILL** occur.

DO NOT impose heavy loads on array frame. Material damage **WILL** occur.

Features

This array has been designed in a "T" orientation to be able to accommodate the VEX™ robot on which it will be mounted.

Vibration isolation is also incorporated in this half size array. The same techniques used in vibration damping for the Tetrahedral Collapsible array are used in this T-Base array. See the previous array feature section for details.

This array also features microphone to microphone distance adjustability. This array can accommodate several microphone types due to the fact that the microphone to microphone distances can be fine tuned. The mounting methods by which the microphones can be attached are also versatile.

Assembly

Major components of tetrahedral array come pre-assembled. Only small adjustments of rods are required to operate array.

Adjustments of rods can be viewed in the exploded assembly drawings in the assembly manual.

If complete disassembly and reassembly is required see schematic layout in the assembly manual.

Operation

The main base of the array has been designed to bolt to the adapter plate provided for attachment to the VEX™ robot.

The microphone adjustability is identical to the Tetrahedral Collapsible array. The same microphone mounting boards and adapter plates are used.

Maintenance/ Adjustment

T-Base Array comes preassembled with main adjustments already in place. For fine-tuning of microphone sensors all extension rods, microphone plates and adapter plates are adjustable using a 5/64" Allen wrench for setscrews.

For maintenance purposes follow these key points:

6.) T-Base array is mainly maintenance free.
7.) Check periodically for signs of wear and tear and replace any damaged or worn components.
8.) Check floating bolt torque to ensure complete damping characteristics on array.

Tetrahedral Array Assembly Views

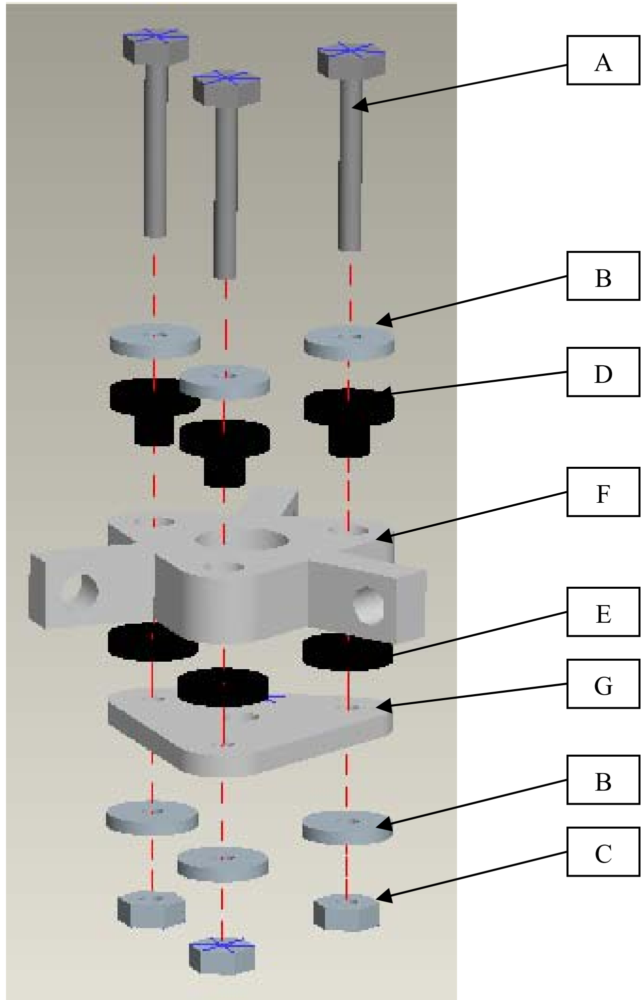
Bill of Materials (Tetrahedral Array Frame)

Part	Designation #	Quantity
2” x ¼” OD Bolt	A	3
¼” Steel Washer	B	6
¼” Steel Nut	C	3
Sorbothane® Bushing	D	3
Sorbothane® Washer	E	3
Main Base UHMW-PE	F	1
RDS Adapter Plate	G	1
Threaded Brass Inserts	H	8
1” Nylon Standoffs	I	2
Motor Adapter Plate	J	2
Interior Slider Link	K	1
6” Lead Screw	L	1
Z2684X-V Stepper Motor	M	1
Igus® Clevis Joint	N	9
Igus® Clevis Pin	O	9
Igus® Clevis Clip	P	9
Acrylic 90º Connector Link	Q	3
Connector Rod	R	3
Extension Rod	S	3
Slider Guide Pin	T	1
Exterior Slider Link	U	1
Center Shaft	V	1
Acrylic Microphone Mount (Bottom)	W	3
Acrylic Microphone Mount (Top)	X	1
8-32 UNC Set Screw	Y	16
Optional Acrylic Microphone Adapter Plate	Z	4

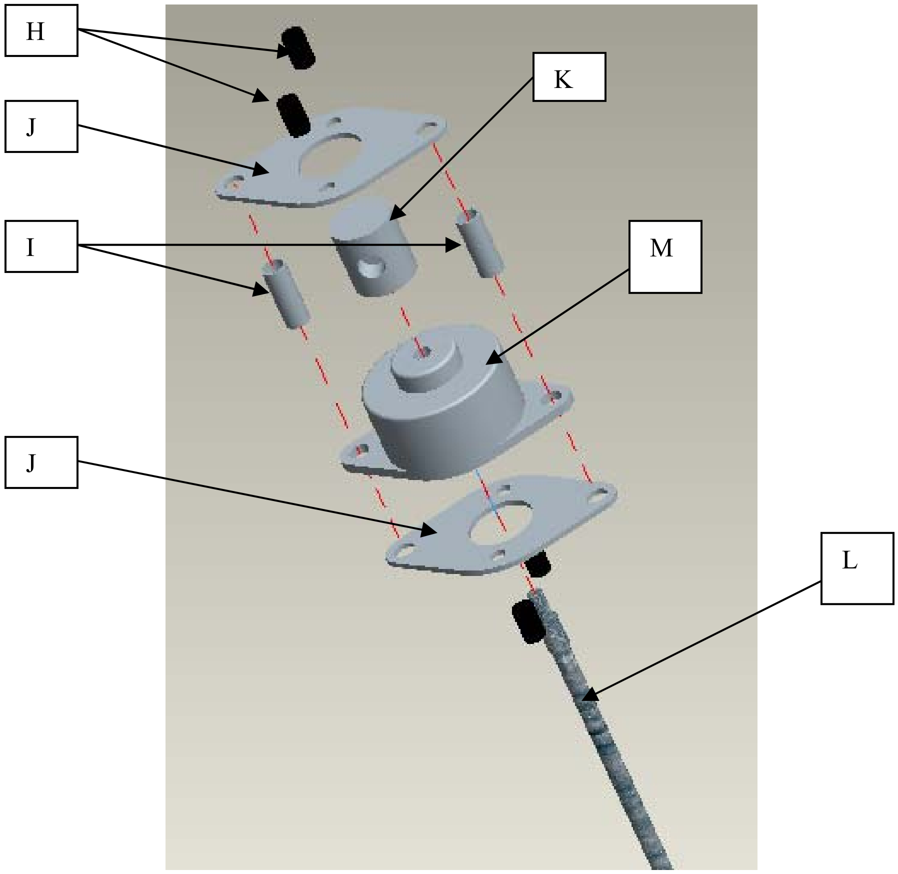
Specifications

Part	Specification	Type
Floating Bolt	5 in-lbs	Torque
Slide Track	4 inches	Stroke Length
Stepper Motor	HSI Z2684X-V	Part #
Guide Pin	3/16”	Major Diameter
Set Screws	UNC 8-32	Thread Pitch
Microphone Mount Bolts	UNC 8-32 Machine Screw	Screw Type and Thread Pitch
Microphone Adapter Mount Bolts	UNC 5-40 Machine Screw	Screw Type and Thread Pitch
Microphone Adapter Plate Standoffs	¼” Nylon	Length and Material
Extension and connecting Rod	½” ABS	Major Diameter
Center Shaft	0.99” Hollow UHMW-PE	Major Diameter
Center Shaft	0.45” Hollow UHMW-PE	Inner Diameter
Motor Adapter Plate Screws & Nuts	UNC 4-40 Machine Screws and Nuts	Thread Pitch

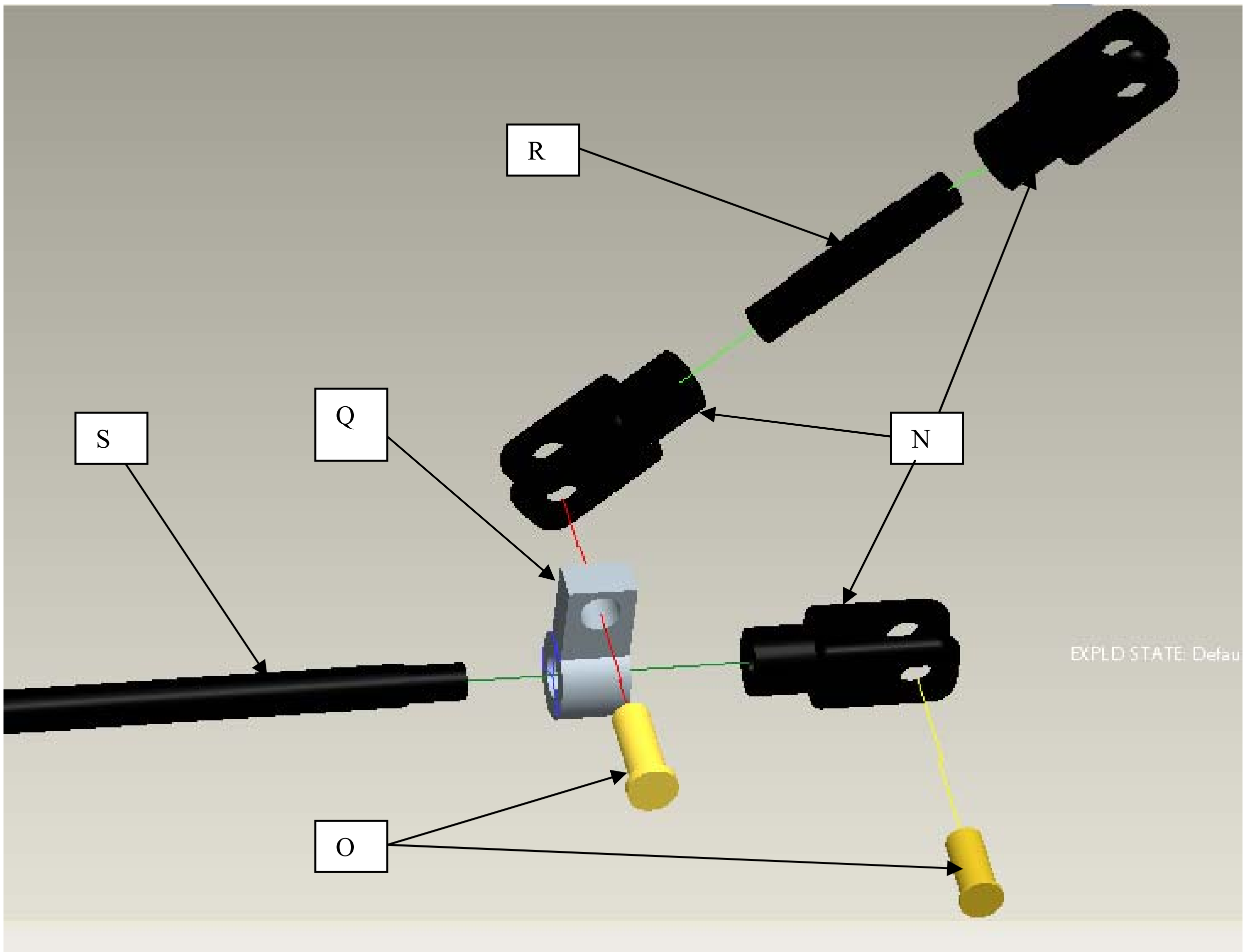
Main Base Assembly



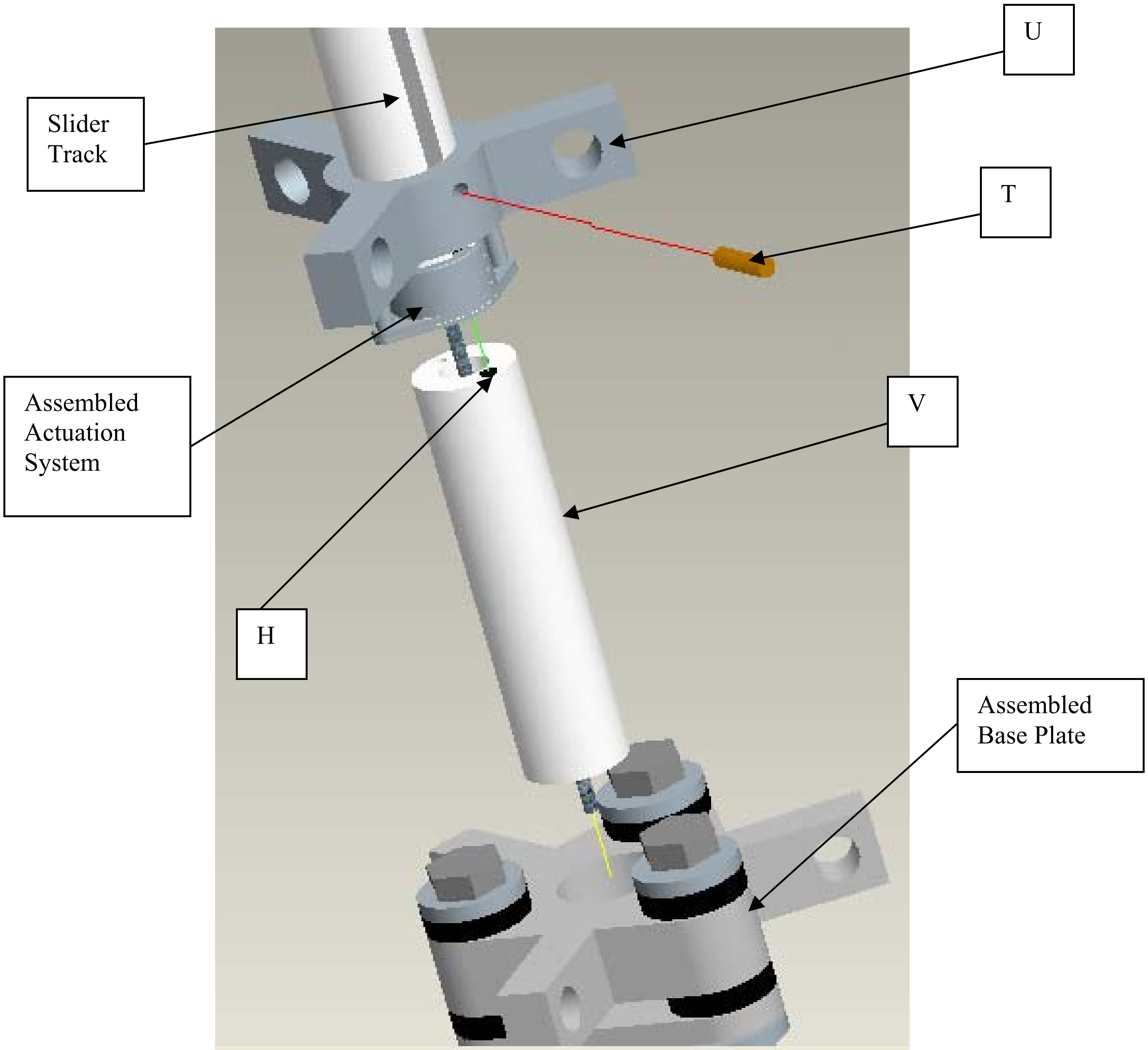
Actuation System Assembly



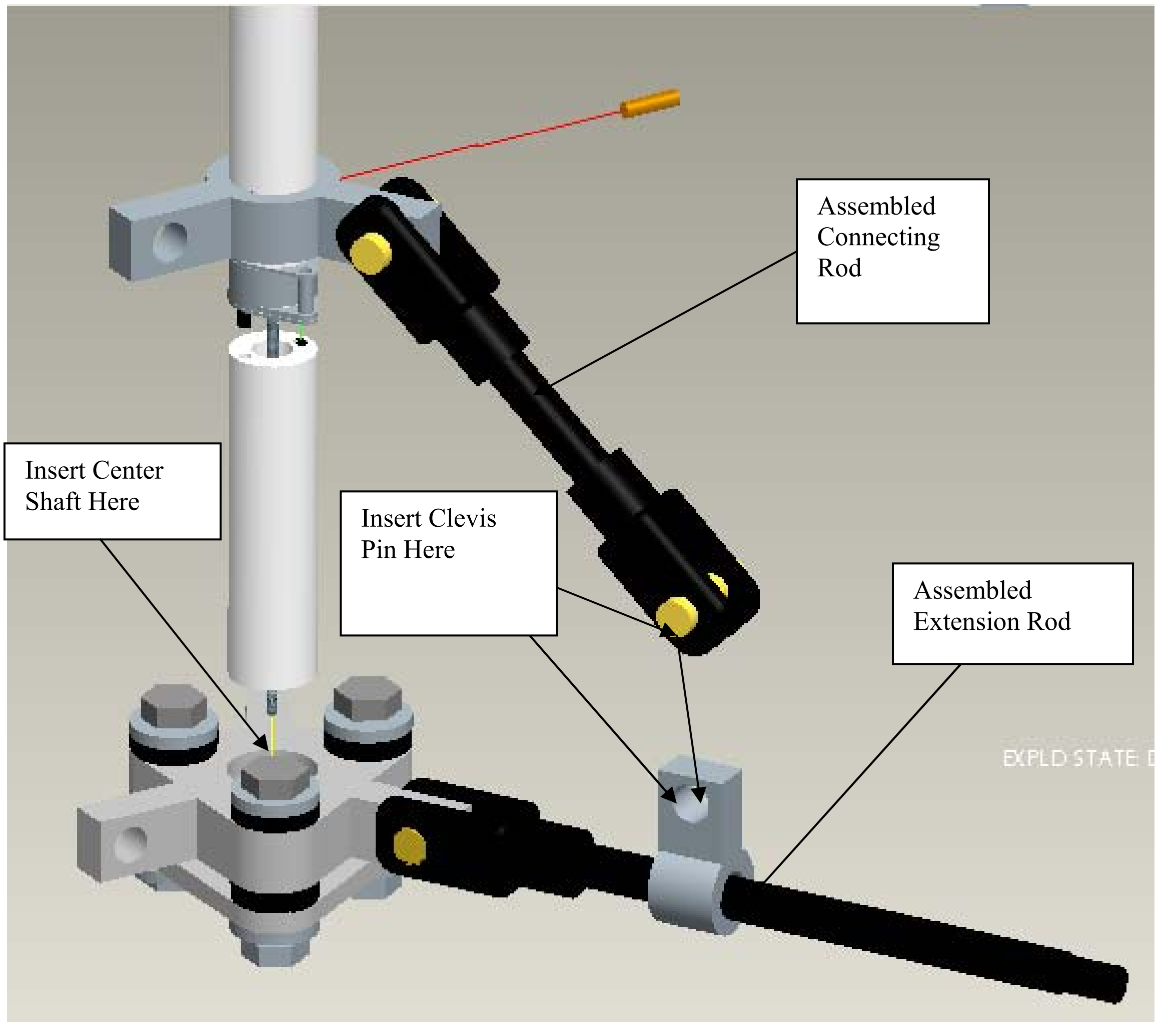
Connecting and Extension Rod Assembly



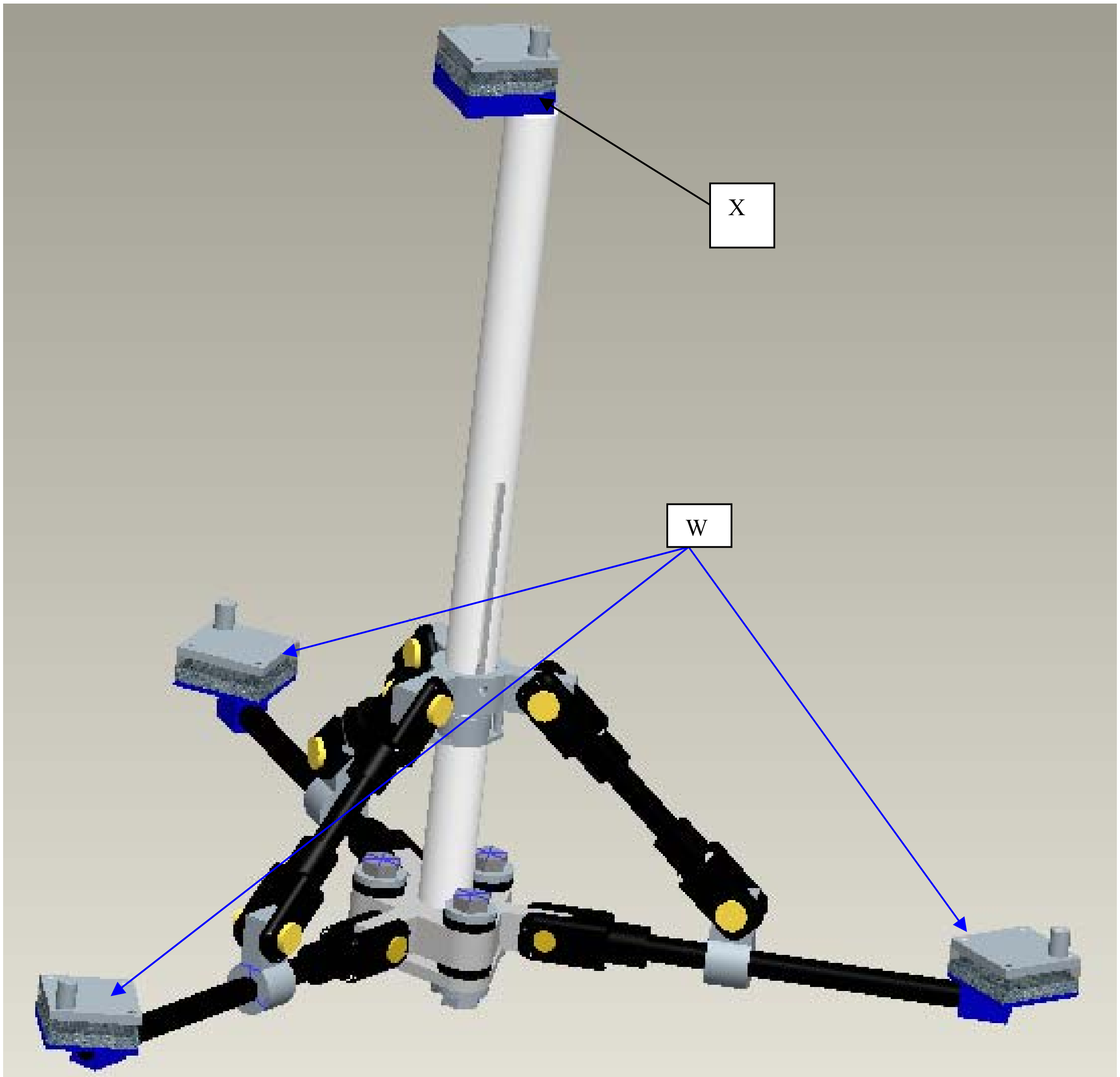
Center Shaft and Slider Track Assembly



Rod to Base Assembly



Completed Assembly



T-Base Array Assembly View

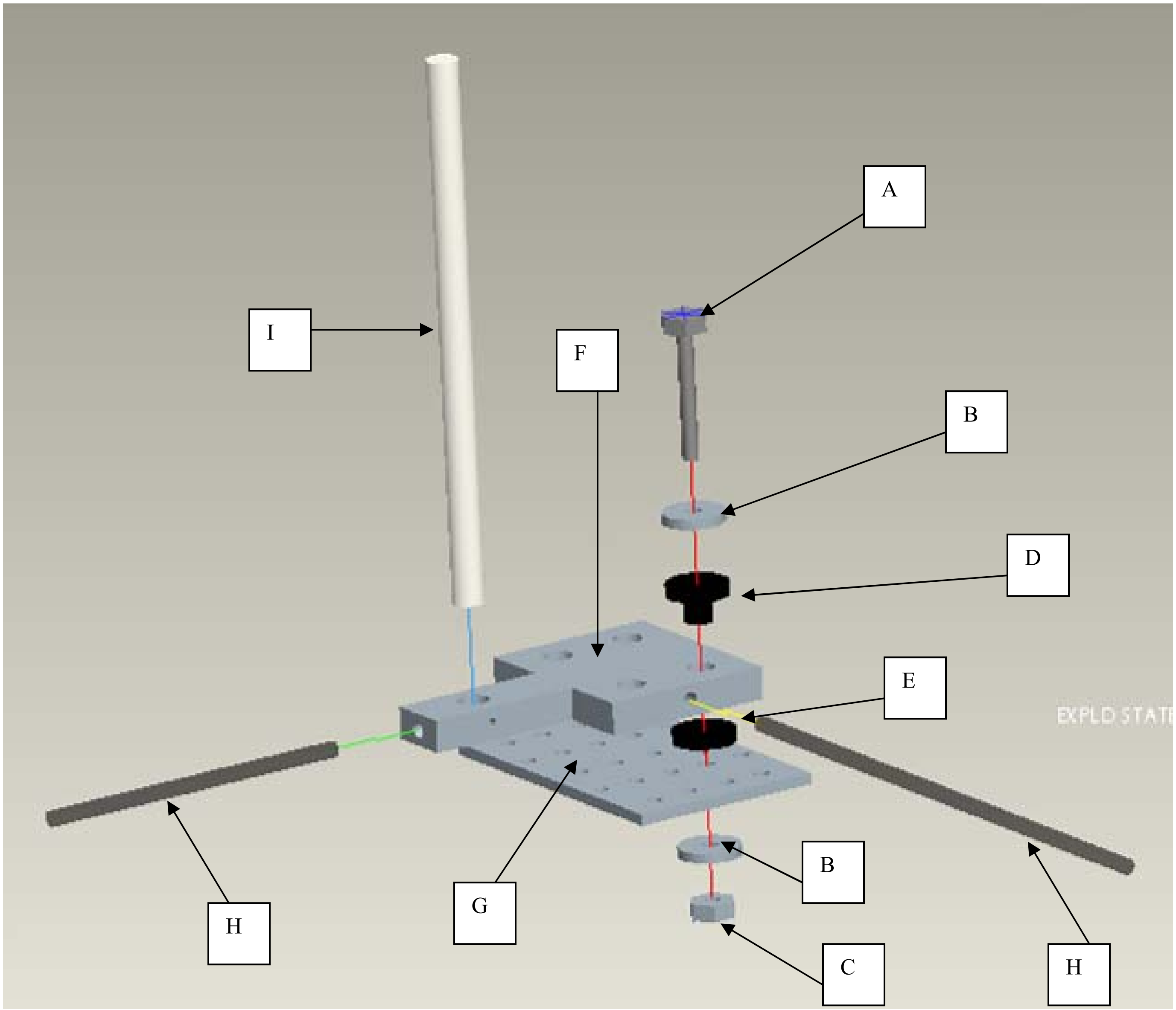
Bill of Materials (T-Base Array Frame)

Part	Designation #	Quantity
1.5” x ¼” OD Bolt	A	4
¼” Steel Washer	B	8
¼” Steel Nut	C	4
Sorbothane® Bushing	D	4
Sorbothane® Washer	E	4
T-Base UHMW-PE	F	1
VEX™ Robot Adapter Plate	G	1
Extension Rod	H	2
Center Shaft	I	1
Acrylic Microphone Mount (Bottom)	J	3
Acrylic Microphone Mount (Top)	K	1
8-32 UNC Set Screw	L	8
Optional Acrylic Microphone Adapter Plate	M	4

Specifications

Part	Specification	Type
Floating Bolt	5 in-lbs	Torque
Set Screws	UNC 8-32	Thread Pitch
Microphone Mount Bolts	UNC 8-32 Machine Screw	Screw Type and Thread Pitch
Microphone Adapter Mount Bolts	UNC 5-40 Machine Screw	Screw Type and Thread Pitch
Microphone Adapter Plate Standoffs	¼” Nylon	Length and Material
Extension and connecting Rod	0.375” ABS	Major Diameter
Center Shaft	0.5” UHMW-PE	Major Diameter
Motor Adapter Plate Screws & Nuts	UNC 4-40 Machine Screws and Nuts	Thread Pitch

T-Base Assembly



Completed Assembly